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Harry T. Woolson, President David Beecroft, Treasurer

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Norman G. Shidle, Executive Editor

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About Authors

• **B. B. Bachman** *joined Autocar back in 1905 when that company made passenger cars exclusively. Since then he has been successively assistant engineer, chief engineer and vice-president in charge of engineering. He was elected to the latter post, which he now holds, in 1929. Active in SAE work since he joined the Society in 1910, he was president in 1922. It was during his administration that the first production session of the SAE was held. He was chairman of the Society's Standards Committee from 1918 to 1922. As a member of the Truck Standards Division he participated in the formulation of the specifications for military trucks for the Quartermaster Department of the Army and in the design of Class A and Class B military trucks. Mr. Bachman was born in Philadelphia and was schooled there, taking his engineering courses at night at the University. He presented his first paper before the Society at the 1913 Annual Meeting.*

• **Dr. Hugo Eckener** *as a journalist and horticulturist 30 years ago had little faith in the dirigible. In fact he wrote a series of critical articles on the subject that aroused the interest of Count Ferdinand von Zeppelin, pioneer of the dirigibles that bear his name, who called on Dr. Eckener and reasoned with him about the practicality of the airship. Dr. Eckener was convinced and since then has been a major figure in the development and history of the Zeppelin. Today his world and transoceanic flights have established lighter-than-air craft as a formidable factor in the world of transportation. The National Geographic Medal was awarded to him in 1930 and he has been five times winner of the Harmon International Trophy. More recently he was presented with the British Gold Medal for Service to Aeronautical Sciences.*

• **P. M. Heldt** *came to this country shortly after completing his public school education in Germany. Continuing his studies at Highland Park Normal College, Des Moines, Iowa, he was graduated in electrical engineering in 1894. After three*

(Continued on page 34)

Publication Office, 56th and Chestnut Sts., Philadelphia, Pa.; Editorial and Advertising Departments at the headquarters of the Society, 29 West 39th St., New York, N. Y. Western Advertising Office, Room 2-136 General Motors Bldg., Detroit, Mich.

Ocean Air Transportation

By L. C. McCarty, Jr.
Project Engineer, The Glenn L. Martin Co.

THE North Atlantic between Europe and North America is the most active oceanic trade route in the world. In a normal year approximately 100 billion pounds of merchandise may be expected to traverse this route.

Of this amount a total of 75 million pounds, less than 1 per cent, is potential cargo for air transportation, consisting of 9 million pounds of precious merchandise, 28 million pounds of express packages, and 38 million pounds of letters and printed matter.

In addition, approximately 250,000 first-class passengers travel each way across the North Atlantic annually. At current rates this potential gross income of traffic suitable for air travel is, roughly, 250 million dollars annually.

Although it will be many years before any such revenues could be realized, this tremendous potential volume of air traffic has stimulated ingenious and intensive development of various technical solutions for the problem of ocean air transport service by several nations.

Complex as the numerous technical solutions are – involving equipment, weather, operating technique, and routes – the problem is complicated much more by delicate international, political, and economic obstacles. Any predictions, therefore, can only be made with the utmost caution. However, qualified opinion, reinforced by considerable experience, is unanimous that regular commercial oceanic air service will be available shortly.

Progress To Date

At the end of the war, occurred the first mad scramble to be the first to fly the Atlantic. Success in some degree was achieved by Alcock and Brown, the flight of the NC's of the U. S. Navy, and crossings of the British dirigible R-34 under Major Scott. The NC's were specifically designed for routine flights across the Atlantic, the shortage of surface craft being so acute that the Navy planned to deliver these patrol boats by air to their bases in European waters. It is worth noting that the NC's had 4 engines, which feature is considered the minimum today for ocean transports, and that the majority of our most modern hull forms were basically developed from their hull lines.

During a long period of time, no further attempts were made to fly the oceans, then in the period 1927-29 practically every sizable body of water in the world was successfully crossed by air. It is probably correct to say that of all the types of aircraft used, not one was designed for regular ocean flying.

At the time that Pan American Airways began to plan for its South American service, the first major trans-oceanic route, there were no aircraft available which incorporated satisfactory design features. The first flying boats used for scheduled flights over a large body of water were the "Commodores," a commercial modification of the Consolidated XPY-1 patrol boats, and the Sikorsky S-38's. The next noteworthy type was the Sikorsky S-40, a large flying boat specifically designed for Caribbean and South American service. It was the first American four-engine commercial flying boat or airplane.

The experience gained from the foregoing types predicated several important principles relative to requirements for regular trans-oceanic service:

(1) Landplanes proved unsatisfactory due to the inherent lack of seaworthiness in emergencies and the necessity for prepared landing areas. A flying boat is more economic because it needs no prepared landing area. The large flying boat permits greater flexibility of operations since it is seaworthy if forced to

L. C. McCarty, Jr.

IN this paper Mr. McCarty, whose company built the "China Clippers," surveys progress in ocean air transportation with particular application to the part of flying boats in the coming struggle for supremacy in the North Atlantic trade. He reveals comparative specifications of new designs weighing up to 250,000 lb. and discloses important data on landing speeds.

[Condensed from a paper presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1937. Mimeographed copies of the full paper are available at SAE Headquarters.]

Fig. 1 – Ocean Transports – Comparative Data

1. Airplane	Type	Latecore 300	Latecore 521	Sikorsky S-42A	Dornier DO-18
2. Payload Capacity	No.	Mail Only	72 Pass. + Mail	32 Pass. + Mail	Mail Only
3. Gross Weight	Lbs.	50,706	82,475	40,000	20,282
4. Weight Empty	Lbs.	-	-	20,931	11,839
5. Cabin Equip. & Furnishings	Lbs.	24,912	41,628	-	1,334
6. Crew	Lbs.	(6) 1,102	(6) 1,102	(6) 1,102	(4) 705
7. Fuel Required	Lbs.	17,990	30,314	12,787	4,497
8. Oil Required	Lbs.	882	1,322	882	441
9. Payload	Lbs.	5,820	8,109	4,298	1,466
10. Engines	Type	4 Hispano-Suiza 12Nbr.	6 Hispano-Suiza Ybrs.	4 P&W Hornet S1EG	2 Junkers "Juno" V
11. Max. Normal Power	B.H.P.	4 x 641 = 2564	6 x 878 = 5268	4 x 750 = 3000	2 x 592 = 1184
12. At Critical Altitude	Ft.	--	--	--	--
13. Normal Engine Speed	R.P.M.	--	--	--	2,200
14. Span	Ft.-ins.	144-11	161-9	118-2	77-8-1/2
15. Length-Overall	Ft.-ins.	84-8-1/2	103-4	68-0	63-3
16. Height	Ft.-ins.	20-11	29-8	17-4	17-10-1/2
17. Wing Area	Sq.ft.	2,800	3,550	1,340	1,056
18. Wing Loading	Lbs./sq.ft.	18.1	23.22	29.7	19.2
19. Power Loading	Lbs./B.H.P.	19.77	15.65	13.33	17.12
20. Max. Speed at Critical Altitude	M.P.H.	130.5	161.5 at 7540	190 at 5000'	155
21. Landing Speed	M.P.H.	--	--	65.2	55.9
22. Rate of Climb at Sea Level	Ft./min.	--	--	799	--
23. Absolute Ceiling-All Engines	Ft.	15,050	20,600	17,550	14,425
24. Service Ceiling - All Engines	Ft.(R.C.= 100 FPM)	--	--	16,000	--
25. Absolute Ceiling-One Eng. Dead	Ft.	--	--	8,525	--
26. Service Ceiling-One Eng. Dead	Ft.(R.C.= 100 FPM)	--	--	7,500	--
27. Range to dry tanks	Miles	2480	2480	2480	2480
28. Cruising Power (Average)	B.H.P.	(45%)1154	(45%)2375	(45%)1350	(45%)532
29. Cruising Speed at Altitude	M.P.H.	84.0	105.0	124	108.7
30. Flight Time	Hours	29.2	23.7	20.0	22.9
31. Fuel Required	Gallons	2,998	5,052	2,131	--
32. Cruising Fuel Consumption	Gals./hr.	102.7	213	106.6	--
33. Cruising Fuel Consumption	Lbs./hr.	616	1278	639.3	196.2
34. Cruising Fuel Consumption	Gals./mile	1.21	2.036	.859	--
35. Cruising Fuel Consumption	Lbs./hp./hr.	.534	.538	.473	.369
36. Payload Factor (#9x29÷2000)	Ton x Mi./hr.	244.4	426.	266.5	79.7
37. Economic Factor (#9x27/2000x#7)	Ton x Mi./lb.fuel	0.401	0.331	0.419	0.404
38. Economic Factor & Cruising Speed (#37 x #29)	Ton x Mi.²/lb./hr.	33.7	34.8	51.9	43.9
39. Economic Factor (#36 ÷ #28)	Ton x Mi./hp./hr.	0.212	0.179	0.197	0.150
40. G.W. Factor (#20x#3 ÷ 2000)	Ton x M.P.H.	3,310	6,660	3,800	1,571
41. G.W. Factor (#40 ÷ #11)	Ton x Mi./hp./hr.	1.29	1.264	1.267	1.327

alight at some place other than its scheduled terminus due to fog, headwinds, and so on.

(2) Larger, more reliable, and more economical powerplants were necessary.

(3) The airplane had to be multi-engined, preferably having at least four power units.

(4) Navigational means and aids to air navigation had to be improved tremendously. Included in this category are radio compasses, upper-air radio balloon soundings, radio drift bombs, and so on.

(5) New instruments were necessary to provide efficient and accurate flight control and to relieve the crew of all possible duties; under this heading are included automatic pilots, reliable fuel flowmeters, and so on.

(6) Greater airplane efficiencies were necessary and obtainable by cleaner and more efficient hulls, airfoils, nacelles, high lift devices, and so on.

Laboratory and flight research has been accelerated tremendously in the last several years, producing really remarkable performances. Of the important types now in use or under construction for existing and projected routes, without exception they are all monoplanes of clean design and all-metal construction. Without exception they have four or more power units.

Fig. 1 shows detail characteristics of several of the more prominent types of ocean transports including characteristics showing the possibilities at 100,000 lb. and 250,000 lb. gross weight with existing materials, technique, and knowledge, and without any fundamentally new developments.

Reviewing briefly the progress to date in ocean flying, we are now well past the first anniversary of scheduled Transpacific air-mail and passenger service, including the long Alameda-Hawaii leg of 2410 miles which is 400 miles longer than the longest North Atlantic leg on the Southern route, from Bermuda to Fayal. In the South Atlantic numerous schedules during seasonal periods have been completed by rigid airship. Last summer the "Hindenburg" completed ten scheduled round trips across the North Atlantic. Again in the South Atlantic, scheduled mail-only service has been operating intermittently for some time by the French, flying non-stop and by the Germans, with catapulted take-offs at both terminals.

North Atlantic Routes

It is probable that for the next several years the principal airway to Europe will follow the "Southern" route via Bermuda and the Azores. Even larger types of aircraft capable of economically flying the whole distance non-stop will be forced

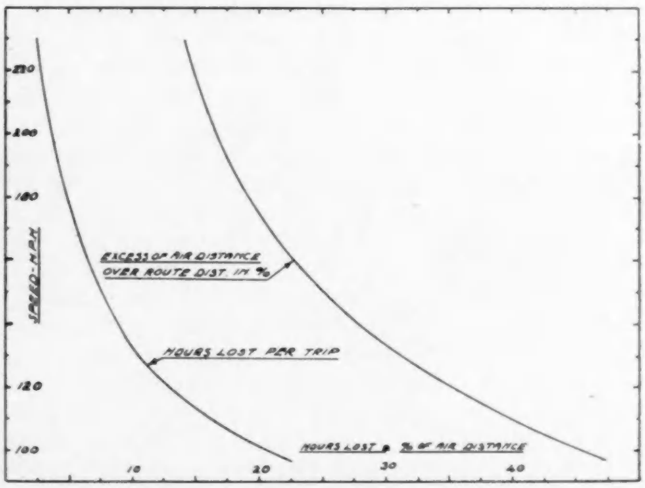

Fig. 2 – Effect of Head Wind at Various Cruising Speeds – Airplane Flying Against a 30 M.P.H. Head Wind for Route Distance of 4500 Miles

Ocean Transports – Comparative Data – *(Continued)*

Martin Model 130		Short "Empire"	Martin Model 156	Dornier DO-20	A	B
46 Pass. + Mail		24 Pass. + 5 Crew	46 Pass. & Crew	?	80 Pass. & Mail	150 Pass. & Mail
52,000		40,500	62,000	110,250	125,000	250,000
25,279	25,279	24,000	29,357	65,047	50,000	109,500
3,338	3,338	3,340	3,413	7,718	5,800	20,000
(7) 1,190	1,190	(5) 1,000	(7) 1,190	(12) 2,205	(10. 1,700	(12) 2,000
13,980	15,400	4,560	14,500	28,100	46,150	58,000
1,090	1,200	400	1,130	2,770	3,150	4,500
7,123	5,593	7,200	12,410	4,410	18,200	56,000
4 Twin Wasp S1A4G		4 Pegasus X-C	4 x Cyclone G-2	8 x Diesels	—	—
4 x 830 = 3320		4 x 790 = 3160	4 x 850 = 3400	8 x 800 = 6400	6,000	12,000
6,000		5,500	5,800	—	—	—
2,400		2,600	2,100	—	—	—
130-0		114-0	157-0	160-8½	200-0	258-0
90-7		88-0	91-6	131-2½	—	—
24-7		31-9-3/4	24-9	31-2½	—	—
2,170		1,500	2,290	4,842	4,030	5,550
23.95		27.0	27.0	22.8	31.0	45.0
15.67		12.68	18.25	17.23	20.8	—
176 at 6000'		200 at 5500	184 at 5800 ft.	180 at ?	—	—
70		73	70	—	—	—
600		950	400	—	—	—
19,500		20,000	16,000	—	—	—
17,400		—	14,000	—	—	—
8,050		—	8,000	—	—	—
5,900		—	—	—	—	—
2480	2480	760	2,480	2,480	* 4,000	* 4,000
(45%) 1495	(60%) 1992	2040	1,800	?	3,710	5,810
122	148	165	140 at 10,000	155	170	210
20.33	16.75	4.61	17.7	16.0	33.6	26.85
2,330	2,567	760	2,417	4,683	7,692	9,667
114.6	153.2	165	136.5	293	228.9	360.
687.6	919.2	990	819.2	1,756	1,373.5	2,160.
.940	1.035	1.00	0.975	1.887	1.923	2.417
.46	.46	0.485	0.455	—	0.37	0.37
434.5	413.5	594.0	869.0	342.0	1548.0	5,880
0.632	0.450	0.601	1.062	0.195	0.788	1.931
77.0	66.6	99.1	148.7	30.2	134.0	405.4
0.291	0.208	0.291	0.487	*	0.417	1.012
4,573		4,050	5,700	9,930	—	—
1.378		1.281	1.676	1.550	—	—

to follow this general course except during the summer season when extensive icing areas and unfavorable weather are not prevalent. Of course, following past practice in both lighter and heavier-than-air craft the route for each flight may deviate widely from the general route in order to properly account for the general circulation about storm areas and other meteorological conditions.

After traffic volume has grown to justify the construction of the larger flying boats, with their attendant increased range, efficiency, and high speeds, the route will shift to a narrow lane following the great circle route. Payloads at these speeds are virtually independent of winds. At economic cruising altitudes of 30,000 ft., above all but the relatively small "peaks" of the weather, the necessity for circuitous detours, which are unavoidable with types limited to moderate altitudes, will be eliminated.

Passenger Comfort and Accommodations

Passenger entertainment, diversion, amusement, and dining facilities on our large surface vessels have reached a high state of luxury and convenience which never can be matched economically in ocean air transportation. Fortunately such luxury on our large super-speed flying boats is unnecessary, and the elaborate facilities, ballrooms, swimming pools, gymnasiums, promenade decks, and so on, may be left ashore where they properly belong. The technical improvements in speed attainable in the larger super air transports permit casting off the burden of extensive passenger "overhead" weight and expense which must be supported by slower forms of ocean transport.

The goal, which is not too far distant in view of the high speeds possible relative to the geography of the world, should be to provide between each important world center and its nearest adjacent center, an overnight sleeper service. Once this goal is achieved, sufficient passenger accommodations will consist of comfortable seats, sleeping facilities, excellent soundproofing, heating and ventilation, convenient lavatory facilities, perhaps individual radios in each compartment, and one or two good meals which need not be of extensive variety.

Until traffic volume grows to warrant the large sizes which will realize such speeds and non-stop ranges, trans-oceanic flying boats probably will continue to take advantage of convenient and economic island bases providing excellent facilities for passenger recreation, diversion, wide variety in cuisine, as well as refueling. Fortunately, some of these island bases, Bermuda in the Atlantic and Hawaii in the Pacific, are important economic centers in themselves and are destined to become important junctions for other trans-oceanic airways in the future.

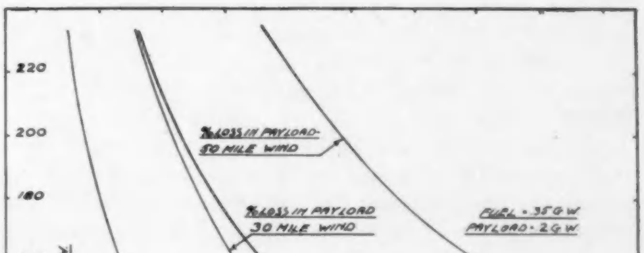

Fig. 3 – Effect of Head Wind on Payload and Extra Fuel Required with Varying Cruising Speed

From this reasoning we cannot anticipate any basic increases in the passenger accommodations on the large flying boats since shorter trip time will result from increasing speeds, even when the longer routes are flown non-stop. Minor refinements of accommodations similar to those provided in the present Martin Ocean Transports are probably sufficient for the future.

Speed

Figs. 2 and 3 show the great importance of high cruising speed when operating against head winds if serious losses and delays are to be avoided. Experience on the Pacific indicates that 30 m.p.h. winds may be expected over large areas, particularly during the winter season, and even greater velocities generally exist over the Atlantic and at high altitudes.

For several reasons we anticipate, therefore, that cruising speeds will increase greatly with 300-325 m.p.h. as a perfectly feasible limit in the not-too-distant future, and with 200-250 m.p.h. as an economic operating speed in the more immediate future:

(1) High-altitude airplanes can be built without any fundamentally new inventions or developments, providing substantial increases in speed and virtual freedom from annoyances and dangers due to weather.

(2) Increases in size within limits permit important aerodynamic refinements resulting in increased range and load-carrying efficiency as illustrated in Fig. 4. From this figure it is apparent that demands for increased speed can be supplied economically.

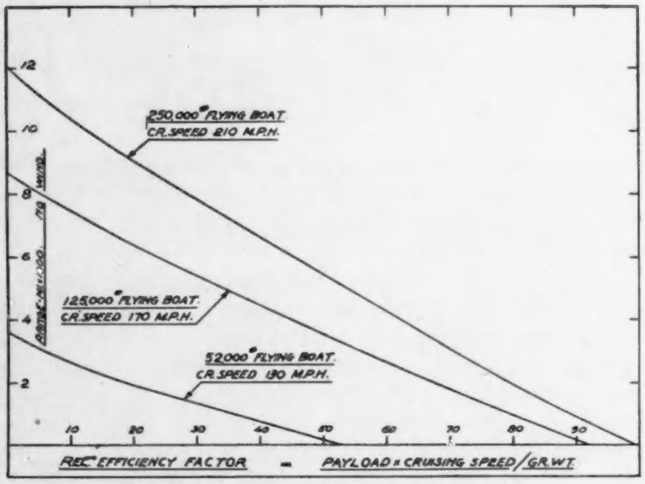

Fig. 4 – Relative Efficiency – Showing Possible Improvements with Increasing Size

(3) Passenger accommodation "weight and space overhead" tends to decrease with speed. Reduced passenger-volume requirements, resulting from increased design speeds, thus introduce a progressive cycle of increasing speed and efficiency.

(4) Competition in traffic procurement inevitably will increase cruising speeds beyond otherwise economic limits since the public is inclined to patronize the swiftest mode of transport even though, in many cases, the saving in time is of no consequence.

(5) Higher speeds increase operating efficiency by allowing more frequent schedules for each unit.

(6) Aviation justifies itself primarily by the fact that it is an instrument providing great speed in proportion to the hazard and expense. Its increasing utility will therefore depend to a very large extent upon increased values in the commodities "speed" and "safety."

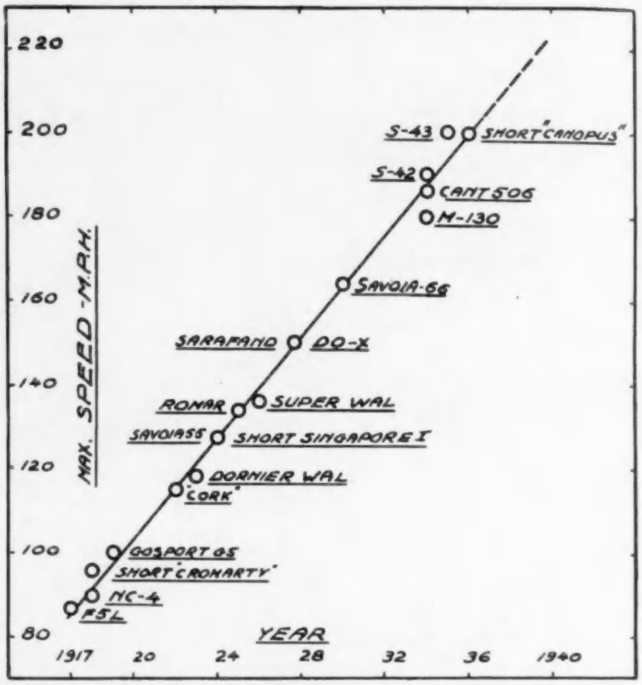

Fig. 5 – Flying Boat Maximum Speeds vs. Year of Construction

Fig. 5 shows an interesting chronological history of flying boat speeds indicating a uniform annual increase of about 6 m.p.h. With the existing state of the aeronautical industry and impending developments in supercharged cabins, this curve probably will increase sharply in the near future as the major advantages of high-altitude flying and scale efficiency are yet to be realized.

Landing Speeds

What is a safe limit for landing speeds? This is a question of such basic importance to future safety, speed and efficiency that it deserves considerable space even in such a brief paper as this.

With no limiting obstacles the safety in landing and taking-off is dependent largely on the impact loads imposed on the structure in the event of bad landings, a bounce or contact with an advancing wave.

From dimensional analysis the following important conclusions are indicated:

(1) The height of free drop increases directly as the linear scale ratio or as weight to the 1/3 power for the same acceleration.

(2) Also for the same acceleration the landing speed increases as the ½ power of the linear scale or as the weight to the 1/6 power.

(3) With the increasing speeds, for a bounce of the same acceleration, the time interval to the subsequent contact with the water is increasing as the linear scale to the ½ power, thus allowing greater time for piloting decision and action.

(4) Since the height of free drop varies with the linear scale, the vertical velocity at contact (for the same acceleration) increases as the linear scale to the ½ power. Extending this reasoning, greater seaworthiness is indicated with the same design accelerations and with the landing speeds increasing as the ½ power of the linear scale.

(5) The physiological limitations of the average pilot in judg-

ing absolute distance above the water as affected by increased landing speeds, sinking speeds, and his height over the water at the instant of landing, are under investigation. Preliminary indications are that, other things being equal, the physiological limitations in this respect have not been approached.

Summarizing, it appears that, as the size increases and without "instrument" landings or the use of power in landing, the average pilot will be able to make landings with lower loads on the structure, less discomfort to the occupants, and with greater seaworthiness at increasing landing speeds. Experience with our Martin Ocean Transports, the largest type built in the United States, is important evidence. Both veteran pilots and those with much less experience have had no difficulty in making consistently good landings from the first attempts. With this airplane the minimum landing speed with full gross weight is 72 m.p.h., with service landing speeds 75 to 80 m.p.h. The height of the pilot's eye above the water at the instant of landing is 15.3 ft. Impact model tests conducted by the Martin organization are conclusive of the validity of this conception of rationalized landing speeds.

As an example, applying these principles, a landing speed of 75 m.p.h. for a 50,000-lb. flying boat would indicate landing speeds of 98 m.p.h. for 250,000-lb. gross weight and 39 m.p.h. for 1000-lb. gross weight.

These facts are presented briefly here to indicate what may be anticipated as the trend in establishing the limitation of the all-important landing speed for the larger trans-oceanic flying boats with reference to seaworthiness, general safety, and the burden upon piloting technique. Emphasis should be placed upon the fact that, for equal safety and for the same size, the

Fig. 6 – Chances of Successful Completion of Any Flight Without Forced Landing Due to Powerplant Failure – Assumption: One Engine Fails Each 531 Hr. of Engine Operation in Worst Case

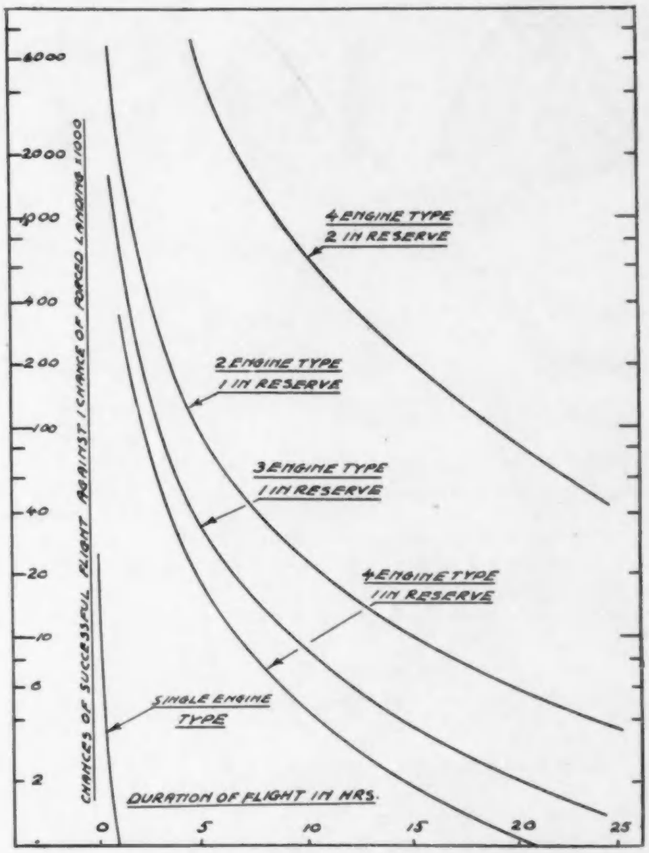

Fig. 7 – Chances of Successful Completion of Any Flight Without Forced Landing Due to Powerplant Failure – Assumption: One Engine Fails Each 1200 Hr. of Engine Operation in Normal Case

safe landing speed is greatly dependent upon controllability and other related hydrodynamic and aerodynamic characteristics. For this reason the landing-speed limit logically should be determined for a particular size from basic considerations as outlined here but modified by the judgment of one or more pilots of wide general experience.

Safety

Of primary interest in connection with the problem of safety is the probability of forced landing due to mechanical failure of the power units. Fig. 6 shows the approximate chances of a forced landing with various types using data from one of the worst six-month periods on domestic transport operations. In this case one mechanical powerplant failure for each 531 hr. of engine operation was experienced. Fig. 7 shows the tremendous increase in safety from forced landing for the multiengine type when the frequency of failure per engine is extended to 1200 hr. which is reasonable for the type of maintenance to be anticipated for trans-ocean service. With a good four-engine boat for such operations the chances of forced landing on each of the long flights is of the order of 1 to 100,000, and this ratio will be increased greatly in the future with six or more engines.

Concluding Remarks

Exactly when the future possibilities briefly outlined in this paper will be realized is difficult to forecast. The time may be accelerated or retarded by military, economic, and political factors. However, it is clear that the future of trans-oceanic transport is assured technically and is destined to be of increasing utility in international trade.

1937 Summer

Tuesday, May 4

8:30 P. M. Auditorium

Aircraft Powerplants

ARTHUR NUTT, Chairman

Trend of Air-Cooled Aero Engines in the Next Five Years – A. H. R. FEDDEN, Bristol Aeroplane Co., Ltd. (To be presented by W. M. EVANS).

Powerplant Trends – G. J. MEAD, United Aircraft Corp. (To be presented by L. S. HOBBS).

Wednesday, May 5

9:30 A. M. Auditorium

Cab-over-Engine Trucks

J. M. ORR, Chairman

Cab-over-Engine Trucks – Their Place in Transportation – PIERRE SCHON, General Motors Truck Co.

Cab-over-Engine Trucks – Their Advance in Design – A. M. WOLF, consulting engineer.

Maintenance of Cab-over-Engine Trucks vs. Conventional Trucks – ROBERT CASS, White Motor Co.

9:30 A. M. Ball Room

Aircraft Radio Shielding

A. L. BEALL, Chairman

Electrical Character of the Spark Discharge of Automotive Ignition Systems – M. F. PETERS, G. F. BLACKBURN and P. T. HANNEN, National Bureau of Standards.

Radio Shielding – H. E. GRAY, radio engineer, American Airlines.

An Investigation of Mica Spark Plugs – M. F. PETERS, H. K. KING, and J. P. BOSTON, National Bureau of Standards.

2:00 P. M. Auditorium

Aircraft Engines

R. N. DUBOIS, Chairman

The In-Line Air-Cooled Aircraft Engine – A. T. GREGORY, Ranger Engineering Corp.

Altitude and Other Variables Affecting Flame Speed in the Otto-Cycle Engine – C. L. BOUCHARD, C. F. TAYLOR and E. S. TAYLOR, Massachusetts Institute of Technology.

8:30 P. M. Auditorium

Oil Temperature Control

F. L. FAULKNER, Chairman

Crankcase Oil Temperature Control – E. W. TEMPLIN, Los Angeles Department of Water and Power.

Thursday, May 6

9:30 A. M. Auditorium

Lubricants

C. H. BAXLEY, Chairman

High Oiliness – Low Wear? – G. L. NEELY, Standard Oil Co. of Calif.

High Pressure Viscosity as an Explanation of Apparent Oiliness – H. A. EVERETT, Pennsylvania State College.

9:30 A. M. Ball Room

Vehicle Performance

S. JOHNSON, JR., Chairman

Fundamentals of Vehicle Performance – M. C. HORINE, Mack Manufacturing Corp.

2:00 P. M. Field Day

8:20 P. M. Auditorium

Business Session

PRESIDENT H. T. WOOLSON, in the Chair

8:30 P. M. Auditorium

Safety

P. G. HOFFMAN, Chairman

Safety in Car Design – J. H. HUNT, General Motors Corp.

Friday, May 7

9:30 A. M. Auditorium

Gadgets

A. G. MARSHALL, Chairman

A Springless Bouncing Pin Indicator – EARL BARTHOLOMEW and CLEVELAND WALCUTT, Ethyl Gasoline Corp.

The Sunbury Knock Indicator – E. S. L. BEALE and RICHARD STANSFIELD, Anglo-Iranian Oil Co. (To be presented by J. R. SABINA).

Spark Advance Indicator – J. R. MACGREGOR and K. R. ELDREDGE, Standard Oil Co. of Calif.

A Spark Advance Indicator for Road Test Use – J. B. MACAULEY, GILBERT WAY and SIDNEY OLDBERG, Chrysler Corp.

Automatic Speed-Load Dynamometer Control – J. R. MACGREGOR and L. T. FOLSOM, Standard Oil Co. of Calif.

Valve Gear and Crankshaft Vibration Studies with Cathode Ray Oscillograph – MAX M. ROENSCH, MAYNARD YEASTING and SIDNEY OLDBERG, Chrysler Corp.

Meeting - May 4-9

Friday, May 7 (Continued)

9:30 A. M. Ball Room

Trailers

G. L. McCain, Chairman

What the Trailer Means to the Car Manufacturer – James H. Booth, Buick Motor Co.

Report of Trailer Hitch Standardization – A. G. Herreshoff, Chairman, SAE Standards Subdivision on Tourist Trailer Couplings.

An informal exhibit of various types and makes of trailers is planned in connection with the Trailer Session.

8:30 P. M. Auditorium

Social Evening

Saturday, May 8

9:30 A. M. Auditorium

Hypoid Gears

Ernest Wooler, Chairman

Hypoid Gears, Axles and Lubricants – W. A. Witham, Gleason Works.

Need for Simplifying Recommendations of Transmission and Rear Axle Lubricants – C. M. Larson, Sinclair Refining Co.

9:30 A. M. Ball Room

Diesel Engines

A. W. Pope, Jr., Chairman

Report of Volunteer Group for Compression Ignition Fuel Research – T. B. Rendel, Shell Petroleum Corp.

Behavior of High and Low Cetane Diesel Fuels – R. A. Rose and G. C. Wilson, University of Wisconsin. (Presented as specially prepared discussion).

Development of the Murphy Diesel Engine – M. J. Murphy, Murphy Diesel Co., Ltd.

8:30 P. M. Auditorium

Body Design

L. L. Williams, Chairman

Artistic Streamlining Against the Wind Tunnel – Alexis de Sakhnoffsky, engineering stylist.

Function in Modern Styling – Frederic A. Seljé, Chrysler Corp.

Sunday, May 9

9:30 A. M. Auditorium

Diesel Engines

T. B. Rendel, Chairman

Recent Trends and Developments in European Automotive Diesel Engine Design – H. R. Ricardo and J. H. Pitchford, Ricardo & Co. (To be presented by Mr. Pitchford).

Diesel Streamliners – Operating and Maintenance Problems and Economies – F. J. Jumper, Union Pacific Lines.

Sports and Social Events

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Tuesday, May 4

Morning and Afternoon – Men's Golf

Wednesday, May 5

Morning – Ladies' Golf
10:00 A. M. – Ladies' Bridge
Afternoon – Men's Golf
9:30 P. M. – Dancing

Thursday, May 6

Morning – Ladies' Golf
10:00 A. M. – Ladies' Bridge
Afternoon – Men's Golf
2:00 P. M. – Field Day
8:30 P. M. – Keno for the Ladies
9:30 P. M. – Dancing

Friday, May 7

Morning – Ladies' Golf
10:00 A. M. – Ladies' Bridge
Afternoon – Men's Golf
8:30 P. M. – Social Evening

Saturday, May 8

Morning – Ladies' Golf
10:00 A. M. – Ladies' Bridge
Afternoon – Men's Golf
9:30 P. M. – Dancing

Public Utilities Fleet Meeting

Four contributors to the success of the Utilities Dinner. Baltimore Section Chairman W. H. Beck *(with paper)* checks final arrangements with Secretary L. W. Shank *(left)*, Vice-Chairman E. W. Jahn *(next)* and Treasurer R. C. Hall *(right)*. Mr. Hall presided at the dinner. These Baltimore officers closely cooperated with the T & M Activity in making the entire meeting an outstanding event.

"An extremely successful experiment."

SAE Vice-President John M. Orr's estimate of the first SAE Regional Transportation and Maintenance Meeting devoted to Public Utility Fleet Operation was more than borne out by the enthusiasm of some 250 engineers and executives who gathered in Baltimore, April 15 and 16 to participate in the sessions. Though termed a regional meeting, public utility fleet operators came from 19 states and Canada. Colorado, Florida and Quebec were the most distant points represented.

Everybody said what he thought throughout the four technical sessions, the safety luncheon and the utilities dinner which made up the program. Circumlocution was noticeably absent. On a wide variety of topics, facts were flung boldly out; experience-borne opinions were voiced frankly.

The dinner speaker, a public utility executive himself, said that utility executives haven't known much about the cost of operating their automotive equipment, but that one of their biggest jobs now is to operate motor-vehicles because the Government has become deeply interested and demands an accounting for every mile of operation, every day.

It took the depression to awaken many utility executives to the fact that their automobiles were costing them a lot of money, another speaker said – and urged that the SAE set up a yardstick for comparing performance of vehicles in regular operation.

At one session several important operators admitted recent conversion to the oil-filter idea. At another, a leading operating engineer said as regards use of aluminum alloy in truck body construction: "Where the gross vehicle weight would remain within the desired limits using a steel body, it seems questionable whether the extra expense would be warranted."

Direction signals, said the safety luncheon speaker, are considered as courtesy gadgets by many people and offer a false sense of security. Argument arose at another session as to whether gasoline or taxes constitute the largest single item of expense in operating an automobile.

. . . And protagonists of the practice of not changing motor oil – merely replenishing instead – laid down figures showing decreased maintenance costs as well as decreased oil consumption to back up their views on this much-debated subject.

From start to finish the two-day session bristled with facts, arguments, ideas – INTEREST.

Conceived by the Transportation and Maintenance Activity, the success of the meeting was the result of cooperative planning by the Activity and the Baltimore Section, which was host to the visiting members and guests. Each Baltimore Section officer was active on the program. Chairman W. H. Beck presided at the afternoon session of the opening day; Vice-Chairman E. W. Jahn acted as Baltimore representative of the T & M Activity and presented a paper at the Economy Symposium; Treasurer R. C. Hall was dinner chairman; and Secretary L. W. Shank was active in making arrangements.

A solo by Miss Mary Gill, accompanied by John Klein of the Baltimore Section, opened the dinner program.

Two Speakers at Utilities Dinner

Toastmaster at the utilities dinner was A. W. Morton, general manager, American Hammered Piston Ring Division of the Koppers Co., who introduced the two speakers, J. G. Holtzclaw, president of the Virginia Electric & Power Co., and vice-president of the Edison Electric Institute, and SAE Secretary and General Manager John A. C. Warner.

"The automotive industry and the electric power industry are twins," declared Mr. Holtzclaw in opening his talk on "How the SAE Can Be Helpful to Public Utility Operators." They were both born at about the same time, he said, and both

Dubbed "Successful Experiment"

have had a great part in advancing civilization. "But," he asked, "would we not be better off if we didn't have them, or radio either? They generate hatred of those who have not toward those who have."

The public utility industry as it is today, he continued, would be impossible without an automobile. The utilities are changing rapidly, and the SAE can be helpful by keeping automotive equipment so that it can be used as an efficient tool. It can help in developing proper maintenance methods and it can aid in keeping the utilities' transportation costs down, he declared.

Many utility executives, he continued, have not known very much about the cost of operating their automotive equipment; they have known the annual costs, but that is all. Now, he said, that the Government has become interested, it appears that one of the utilities' biggest jobs is to operate motor-vehicles. They have to account for every mile of operation every day, he stated, adding that economy has become an all-important factor.

Opportunity for Cooperation

Mr. Holtzclaw sees an opportunity for cooperation. The manufacturer, he said, develops his theories and gives the fleet operator something to work with, while the operator in return provides the manufacturer with a proving ground to test out his theories.

Mr. Warner emphasized the importance of the public utility fleet-supervisors' job. Most executives in the business, he said, are trained to distribute gas, electricity or furnish means of communication and know nothing about operating a fleet of motor-vehicles; they leave that up to their fleet supervisors. He showed how the SAE, through its open-forum meetings, its standardization and research work, and its publications can be of real service in helping fleet supervisors in their work.

"This meeting," he said, "exemplifies SAE advancement. It is the first of its kind to be held and one that seems likely to be repeated regularly, because it has a vital usefulness to a live part of the industry. Probably no branch of motor-vehicle operation has as many problems or changes so rapidly as do public utility fleets."

A few words of welcome from Vice-President Orr of the Transportation and Maintenance Activity opened the meeting's first session. This event, he said, is the first SAE T & M Regional Meeting to be devoted to the problems of one industry. He predicted active interest as he looked over the 150 engineers already assembled. Mr. Orr then turned the gavel over to Adrian Hughes of the Baltimore Transit Co., chairman of the session, who in turn introduced Prof. James I. Clower of Virginia Polytechnic Institute.

"I am afraid some operators take it for granted that correct lubrication is assured when the most suitable oil has been selected," remarked Professor Clower at the start of his paper, "Oil Filters in Public Utility Operation." Proper care of the lubricant while in service is essential to insure minimum cost per unit for safe and dependable operation, he contended.

Stating that oil technologists and other experts have known for many years that lubricating oils do not wear out, but simply become contaminated with various impurities, Professor Clower added that their removal renders the oil suitable for further use. A good oil filter, he said, performs this service to the extent that when it is used the period of oil draining can be tripled or quadrupled. Many large fleet owners are operating from three to six or more thousand miles between drains, and some never drain except for the purpose of changing from summer to winter grade and vice versa, he stated.

In summing up the possible benefits accruing from the use of oil filters Professor Clower expressed his belief that use of one of the more efficient types will greatly extend the period between oil changes; generally reduce piston, ring and bore wear and valve and ring sticking. Some of the present-day filters, he added, maintain the acidity of the oil at a low value, thereby minimizing corrosive effects, especially on the newer copper-lead and cadmium-silver bearings. While the day is not here now, he believes it will come when oil will not be changed except to meet seasonal demands.

Renewing of the filter cartridge, the author noted, is dependent on the characteristic of the oil being used, the condition of the engine, operating and climatic conditions and the character and capacity of the filtering element itself. Under normal conditions, he believes, it should be capable of rendering effective filtration for at least 2000 miles, though many under severe conditions perform satisfactorily for from four to five times this distance; the average probably being from 3000 to 4000 miles.

In bringing his paper to a close Professor Clower emphasized the need for a simple test for determining when an oil is un-

John M. Orr, SAE vice-president representing the Transportation and Maintenance Activity, welcomed the public utility fleet operators to the first T & M meeting to be devoted to the problems of one industry.

suitable for further service. He pointed out that there has not yet been developed a scientific test or series of tests capable of doing this. As a result, because no one feels justified in assuming the risk of a costly repair job for the comparatively insignificant cost of an oil or filter-element change, millions of miles of oil life are annually being run to the sewer. This, he declared, is not only important because of the cost to motor-vehicle owners and operators, but also from the standpoint of the conservation of one of our country's most essential natural resources.

F. K. Glynn, of American Telephone & Telegraph Co., was the first to answer Chairman Hughes' call for comments. Prefacing his remarks with the statement that he believes in filters, he expressed his belief that operating temperatures have considerable effect on the development of sludge.

He also questioned the detrimental effect of sludge, noting that during a cold winter, when oil screens tend to clog, some operators bypass the oil without experiencing serious results.

Until recently "filters have been the bunk," said E. J. Fraser, Baltimore Transit Co. Now that they are larger and the pressure of the oil through them has been reduced they are more effective, he added. However, he feels that the oil filter takes out of the oil some of the elements that belong there.

E. W. Jahn, Consolidated Gas, Electric Light and Power Co., said that sludge doesn't seem to form where it does damage, noting also that sludging occurs mostly at low operating temperatures.

Admitting that he was originally opposed to oil filters, Robert Cass, White Motor Co., said he is now convinced of their value. Mr. Cass suggested that standardization should be undertaken to recommend a standard size of filter for each size of engine, recommending that this be based, in general, on the straining area.

J. R. North, Commonwealth & Southern Corp., condemned the use of an abrasive filter element and noted that connections should be full metallic and rigid.

Among others contributing to the discussion were A. W. Morton; C. M. Billings of Gulf Oil; Capt. O. A. Axelson, Columbia Gas & Electric Corp.; James E. Hurn, DeLuxe Products Corp.; and W. C. Bauer, Briggs Clarifier Corp.

Aided by numerous carefully selected slides, T. C. Smith, American Telephone & Telegraph Co., at the afternoon session, traced the development of trucks used by his company for utility transportation, construction and maintenance from the "nineties" to the present. He was introduced by Baltimore Section Chairman W. H. Beck, Sherwood Bros. Inc., who presided at the session.

Accidents Factor in Early Days

"Be Careful of Accidents," a sign painted on the side of a horse-drawn truck pictured in Mr. Smith's first slide showed that safety has been an important factor in utility operation at least as far back as 1896. He noted that it was about 1910 when carefully prepared studies indicated the practicability and economy of using gasoline-driven motor trucks, and he showed examples of the equipment used at that time.

The author then skipped some 25 years, picturing various types of equipment which have recently come into use, and touched on some of the technical problems that these new trucks have developed.

Regarding the use of aluminum alloy in body construction, he said that it is warranted in cases where its use permits keeping the gross vehicle weight within permissible limits for a particular size of chassis, but "where the gross vehicle weight would remain within the desired limits using a steel body, it seems questionable whether the extra expense would be warranted. The low daily mileage has a major effect upon this matter. Utilities should, it seems, not be influenced too much by ton-mile considerations in studying the proper field for the much more expensive lighter weight body materials." Galvannealed steel, he said, is used quite generally for bodies. He noted a number of its advantages over other materials.

Comparative Body Costs

The first cost of one of the small-type bodies, when made of aluminum, is roughly 50 per cent higher than for galvannealed steel, and the first cost of a galvannealed-steel body is approximately 8 per cent higher than for black sheet, he added.

Mr. Smith stressed the problems arising, particularly in the lighter truck units, from the considerable variation in the cab-to-axle dimensions in commercially available chassis. This, he said, results in difficulty for the designer as well as the body manufacturer. He also suggested that the information supplied on chassis drawings, for the body builders, indicate the dimension from the center of the axle to the top of the frame with metal-to-metal contact between frame member and axle housing, as this would assist in planning the wheel-house heights.

In the latter portion of his paper, Mr. Smith described the auxiliary equipment used by a utility fleet, including pole-trailers, truck-engine-driven centrifugal pumps, winches, earth boring machinery and ladder platforms. A short motion picture showed the latter equipment in use.

Elmer J. Graham, Public Service of Colorado, referred to five and seven-man cabs, which had been described by Mr. Smith, and asked how the truck driver can see to operate the winch. Mr. Smith said that a large window is used and that a periscope is coming into use for this work.

The importance of balance between body life and chassis life was brought out by Mr. Cass. Bodies, he said, should be so constructed as to last as long as the chassis without much maintenance, but not to last too long. On this same subject Captain Axelson said that in his experience cabs are not built in that balance, because one engine and chassis lasts as long as two cabs.

Captain Axelson, and H. O. Mathews, Public Utility Engineering & Service Corp., brought up the subject of standardization – Captain Axelson of bodies for utility trucks and Mr. Mathews of auxiliary power equipment. Jean Y. Ray, Virginia Electric & Power Co., believes that there is need for standardization but that changes in construction practices are working against it. His own company, he said, is developing a two-man line truck for use in making rural installations.

Mr. North, at that time, presented written discussion on utility trucks with standardized bodies. His company, he said, has also made an extensive study, as a result of which they have developed certain standard body designs employing all-metal construction. These are lighter, stronger and, in many ways, superior to wood or composite wood and steel bodies formerly used. These bodies are mounted low on metal sills and are adaptable to various chassis produced by different manufacturers.

Standard bodies for a number of applications have been constructed, Mr. North continued, and in his discussion described the light line-construction-and-maintenance truck, the heavy truck for this type of service, a pole-hole-digger truck and a general service truck. Considerable savings over the cost of non-standard trucks are experienced, he added.

Stating that he had noticed a metal safety tread on some of the trucks pictured in Mr. Smith's slides, Randolph Whitfield, Georgia Power Co., asked if it did not tend to get slippery when greasy. John M. Orr answered this question by stating that his company has used this type of floor for three years without experiencing trouble. Mr. Orr also suggested waterproofing tool compartments on trucks.

Speaking from a manufacturer's point of view, J. H. Mack, Fargo Motor Corp., referred back to standardization. He would like very much to see it. When the manufacturing department gives the sales department a price on some special job it is often so high that the salesman is ashamed to quote it, he said, adding that in mass production anything different costs money, and that when it comes to special dimensions the cost mounts rapidly.

A. J. Scaife, the Autocar Co., suggested that manufacturers use cab-to-axle dimensions instead of wheelbase. Uniformity there would help body builders and designers in developing bodies to fit chassis of the various manufacturers, he added.

A. H. Bishop, Autocar Sales & Service Co., took the chair for the Friday morning session. A paper, "Executive Control of Public Utility Fleet Operation," by F. B. Flahive, Columbia Gas & Electric Corp., opened the program. This was followed by W. R. Pollard, of the Georgia Power Co., who spoke on "Efficient and Economical Operation of Automotive Equipment of a Public Utility."

"The depression forced management to watch where every dollar was going, and it was then that many utility executives woke up to the fact that automobiles were costing them a lot of money," Mr. Flahive stated.

Cost-Per-Mile Figures Misleading

Before outlining methods of executive control, which he defined as "plain every-day management," Mr. Flahive suggested that the SAE set up a yardstick for comparing the performance of vehicles in regular operation. He declared that the cost-per-mile figure is totally inadequate because of varying accounting procedures and varying costs of fuels and lubricants – particularly when one car is supplied with oil and gas by retail service stations and the other by the company garage at considerably less cost. Cost-per-mile figures on this basis, he claims, are misleading.

The first system of management discussed, he termed "direct control." Mr. Flahive explained that under this method the parent company sets up a transportation department, headed by a man of ability and automotive experience, which has complete charge of all transportation policies. This department passes on all requisitions for purchase of equipment and supplies, directs operation and maintenance, and, in some cases, actually designs equipment. This method, he said, has the distinct advantage of providing one central authority to assume full responsibility for transportation. The main objection, Mr. Flahive declared, is that it takes too long a time to get decisions from the main office; another that the cost of such a transportation-control organization may well exceed any savings effected by its work.

"Control by analysis and suggestion" was the second method described. In this, the author said, the central office employs an automotive engineer who is qualified to analyze the operations of the fleet from cost and statistical data supplied by the operating companies. The responsibility for operating and maintaining the fleets remains with the operating executives who may or may not act on the suggestions made by the engineer. This method of control, Mr. Flahive said, has the big advantage of being inexpensive, and it is also fairly effective as operating executives are anxious to cut expenses and, in most cases, will act on the suggestions offered. The outstanding objection, he feels, is that the engineer is offering suggestions to the operating executive, who is not an automotive man, and misunderstandings are apt to occur as they do not speak the same language.

At Speakers' Table

Speaker J. G. Holtzclaw *(left)* and Toastmaster A. W. Morton swap stories at the Utilities Dinner. As vice-president of the Edison Electric Institute, Mr. Holtzclaw brought greetings from that organization.

In Mr. Flahive's company the method used is "committee control." This was the third system explained. In it, he said, the board of directors of the parent company appoints a committee consisting of the man from each operating group most directly in charge of motor transportation, with an officer of the company as chairman. Its secretary is an automotive engineer, a man able to furnish technical advice, who also carries on certain activities in conjunction with other committee members. The purpose of this committee, Mr. Flahive explained, is to inquire into the cost of fleet operations and to reduce these costs when possible. As the members are the operators, he continued, they are in a position to apply such remedies without the usual recourse of making recommendations for executive approval. In his company, Mr. Flahive said, the meetings developed into a round-table discussion of such subjects as retirement age of vehicles, use of lighter vehicles, oil changing and oil filters, periodic inspection, and training of drivers. During four years of operation under this method the total expense for operating his company's fleet was reduced 25 per cent, Mr. Flahive reported.

He deplored the use of gadgets, such as dome lights, chromium-plated hardware and fancy instrument panels on commercial cars. They are not needed, he said, and they cost more money.

In the discussion Mr. Orr brought out the fact that in many instances a company pays $1000 for a chassis, but by the time it is equipped for service the body and auxiliary equipment bring the total cost to $4000 or $5000. He said that it is built primarily as a tool, not for mileage economy. He agreed with the speaker that there should be a yardstick for comparing performance of vehicles, and referred to the SAE plans for standardized cost accounting, as a very definite step in this direction.

Mr. Glynn observed that, perhaps, the fleet operator does not have the position he should have in the company. He is an engineer carrying the responsibility of managing considerable capital equipment and spending a lot of money and this responsibility should be accompanied with authority.

Regarding gadgets, Mr. Mack said that it frequently costs more to leave them off than to put them on. If the light cars for fleet operators were built for utility alone, he added, their trade-in value would be much lower. This, he explained, is because these cars must be sold by dealers to the public, and the public wants gadgets.

Automobile an Indispensable Tool

"As far as utility executives are concerned, an automobile or truck is an indispensable tool. When it is understood that such tools cost approximately $600 per year to operate, and that often hundreds of these tools are used by a single utility, they should realize that the management of such a fleet is of sufficient magnitude to require the services of an automotive engineer and several assistants," Mr. Pollard said in his paper. He emphasized that the automotive supervisor should report directly to the management to obtain the best use, and enforce the proper control of equipment without embarrassing some department head. He also stressed the necessity of training mechanics, as many of them are still in what he termed "the screw driver and plier stage."

Selection of proper equipment for each job, he declared, is important as it results in saving the crew's time as well as in lower trucking cost.

To best operate equipment, Mr. Pollard believes that the automotive supervisor should be advised of any changes in methods of operation, construction programs and sales campaigns. He has found it necessary to assign certain cars for personal use, but suggests keeping this number to a minimum and pooling other automobiles for the use of as many employes as possible. In some instances, he said, it is to the best interest of the company to allow employes to use their personal cars on company business. In such case the employe should carry insurance to protect the company and himself, and receive compensation for actual mileage used on company business, he added. His experience indicates that this should be $0.05 per mile for a maximum of 500 miles per month. He mentioned that some companies have found that the use of governors and throttle stops result in economies worth striving for.

All cars and trucks should be inspected and tuned up approximately every 3000 miles, he declared, adding that major repairs should be made as necessary. On the subject of tires, he stated that they should be inspected once a week for inflation. Experiments with tire retreading, he reported, have shown that it costs from 33 per cent to 55 per cent of new-tire price and gives about 75 per cent as long life.

The economical trade-in point for cars and commercial trucks can be found graphically, Mr. Pollard maintained and illustrated the method used by his company.

When Mr. Bishop asked for discussion on Mr. Pollard's paper, Mr. Scaife brought up the question of using a smaller carburetor instead of a throttle stop for the same purpose. It was brought out that this is done on some economy cars.

Use of personally used cars on contract basis sometimes results in the company paying for pleasure jaunts, said Mr. Mathews, who added that this can be eliminated to some extent by reimbursing the employe on a sliding-scale basis.

Extremely popular was the "Safety Luncheon" on Friday. It was presided over by J. R. Sherwood, senior vice-president, Baltimore Safety Council, who introduced the speaker, J. W. Lord, Atlantic Refining Co.

A skilled driver, like a good athlete, must place emphasis on correct form, good technique and use of his eyes, said Mr. Lord. He feels that use of eyes in driving is particularly important and should be carefully studied. Eyes must be trained to take in a broad panorama, he believes, rather than to focus on some one item; to see someone in the road but not to know who he is.

Speaking of directional signals, he said that they are considered "courtesy gadgets" by many, offering a false sense of security. Their use, he declared, involves a physical distraction to the driver and an eye distraction to those supposed to see and read them. He likes them on big trucks, but feels that their use should be augmented by proper positioning of the vehicle on the highway to indicate direction of turn; on the right side for a right turn and in the center to turn left.

With a series of slides he pictured driving conditions that are conducive to accidents, drawn up from accident records of his fleet, illustrating the dangers of passing at intersections, following another car too closely, relying on hand signals without also looking to see what the other fellow is doing, depending too much on other drivers obeying highway-stop rules, and a number of other bad driving habits.

Economy Symposium Concludes Program

The Economy Symposium, at which Captain Axelson, Mr. Jahn and Mr. Ray read papers concluded the two-day meeting which Vice-President Orr summed up as a highly successful experiment.

In a fleet operating 30,000,000 miles per year the difference of one-tenth cent per mile means $30,000, said Captain Axelson, in introducing his paper on "Throttle Stops." Continuing he said that gasoline is the largest single item of expense in operating an automobile and that increasing the miles per gallon by only one mile is equivalent to a saving of more than one tenth of a cent per mile, at average gasoline rates.

In his company he found that by limiting the throttle to about one-third open throttle, light vehicles still had a top speed of 55 to 60 m.p.h., and a pleasing economy in fuel consumption. He told of further economy by increasing the heat efficiency of the engine.

His company, he reported, is now operating more than 900 throttle-stop equipped vehicles, and it is his opinion that these have done more to increase the economy of the fleet than any other single factor. While installed primarily to increase gasoline mileage, which they do to the extent of some 30 per cent, he reported that tires are lasting from 25 to 50 per cent longer, less oil is being used, road failures are less frequent, rear-end troubles have practically been eliminated, accidents have been less severe and maintenance greatly reduced.

In concluding, he stated that for very hilly country and for roads and loads requiring every bit of available power the use of economy models may not be advisable.

John White, of Mack Trucks, a former chairman of the Baltimore Section, presided at the Economy Symposium. He asked for discussion at the close of Captain Axelson's paper.

Mr. Whitfield told that in his company, the Georgia Power Co., the fleet of passenger cars and light trucks is 95 per cent equipped with throttle stops. They have found, he said, that it is best practice to use a one-half throttle setting, which permits 60 m.p.h., and then use a governor to limit the speed to 45 m.p.h., which they consider the maximum for safety. By

(*Continued on page* 52)

Diesel Engines in Trucks

By B. B. Bachman
The Autocar Co.

THE author makes a brief recapitulation of the paper that was presented at the Metropolitan Section of the Society last spring and carries through an analysis of reports that have been received on the operation of some 30 of these units both in the East and on the West Coast.

The discussion covers the general operating characteristics of these units together with a further study of a comparison between two Diesel units and a gasoline unit from the basis of operating cost.

EARLY in 1936, the writer presented a paper to the Metropolitan Section outlining some thoughts on the subject of applying Diesel engines to trucks. In that paper were given the reasons that prompted building such equipment, a general description of it, and the effect on relative dimensions and weight. The relative performance of two of these units, equipped with engines of different make, compared with a somewhat similar gasoline unit was discussed. This discussion was necessarily incomplete due to the time having been too short to develop sufficient data.

General conclusions arising from the experience obtained up to that time were given then and will be repeated here as a basis for amplification in the light of additional experience:

(1) Diesel equipment for a given service will weigh more and cost more than gasoline equipment. Weight can be reduced by using engines more nearly comparative in size and by scaling accessory chassis details.

(2) The basic differences in the engine cycles need recognition in manufacture, service, and operation. These differences are not difficult to learn by competent men in either class.

(3) Although current maintenance of the Diesel trucks does not seem to be greater, there is evidence that it may be necessary to do tune-up work on rings, pistons, and so on, at more frequent intervals and earlier in the operating life which work will affect maintenance costs unfavorably.

(4) Even with such careful preventive measures, the overhaul period of Diesel trucks may arrive at an earlier time.

(5) Fuel consumption can be expected to be approximately 50 per cent less than that of a gasoline engine. The total saving in cost will depend on many other items which render it impossible to make an intelligent general statement.

(6) Oil consumption may be about the same as in gasoline engines, but the difficulty of keeping the oil clean will call for more frequent oil changing and also more frequent filter-element changes.

(7) For best results somewhat different driving is desirable. The Diesel engine should be kept within a relatively limited speed range.

(8) Changeover installations can be made if careful attention is given to the particular case.

The factor of basic importance is whether Diesel equipment can be operated more economically than gasoline equipment. Under favorable and proper operating conditions, the answer is undoubtedly yes but, as such results do not always follow, our purpose herein is to see if we can find why there should be such differences.

A considerable number of the units that we have built have been delivered to customers on the West Coast. This territory apparently has most favorable operating conditions for Diesel equipment, and the general results obtained are more satisfactory than in the East. It is not possible to say exactly why this condition should be but, as it seems to be a matter of fact, it may be permissible to make some guesses:

(1) The limitations on size and weight do not impose such severe restrictions as they do in the East.

From The National Highway Users' Conference analysis of "State Restrictions on Motor-Vehicle Sizes and Weights," we find that California, Nevada, and Oregon do not limit the number of units in a train. Length limits in California are 60 ft.; Nevada, 60 ft.; Oregon, 50 ft.; Utah, 55 ft.; and Washington, 85 ft. Axle-weight limits are: California, 17,000 lb.; Nevada, no limit; Oregon, 16,000 lb. with 17,000 lb. on paved highways; Utah, 18,000 lb.; and Washington, 18,500 lb. Gross-weight limits in California are 68,000 lb.; Nevada, 114,000 lb.; Oregon, 54,000 lb.; Utah, 700 (L plus 40) which will admit about 60,000 lb.; Washington, 68,000 lb. From this data it can be seen that here is a large territory involving long hauls which has more liberal allowances for size and weight than a similar comparable area elsewhere in the country.

(2) The length of haul on the average is greater.

The general character of this district and the increasing demand for transport service for which there is no other agency available create more long haul business than is general elsewhere.

(3) For the foregoing reasons the cost of fuel assumes greater importance than it does in shorter-haul operations.

As the length of haul increases and as the weight of equipment increases, there is a corresponding increase in the total

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1937.]

amount of fuel used and in the cost of the fuel. Although the cost per mile or per ton-mile is generally less for long haul and heavy equipment, the ratio that fuel cost bears to total operating cost is greater. For these reasons, the reduction of this item of operating cost presents a very attractive possibility.

(4) There is a greater differential between the cost of fuel oil and gasoline than in the East.

It is hard to treat the matter of relative cost of fuel oil and gasoline with complete accuracy as conditions change rapidly but, in general, from the most recent information that we have, the price of fuel oil on the Coast is less than 50 per cent of the cost of gasoline and may be as low as one-third of it, whereas in the East it is 80 per cent and rarely less than two-thirds. This differential, naturally, produces a much larger margin of saving with which to balance some of the other unfavorable factors.

(5) There may be some inherent characteristics of the fuel oil obtainable on the Coast that make it more satisfactory.

This item is very questionable, and we realize that, in even mentioning it, we open the door to great criticism. Therefore, we hasten to say that we have no definite data upon which to base such a statement and admit that we are not even qualified to compare specifications critically. Following are the specifications of three fuels that are being used on the Coast with the thought that those who are in position to analyze this subject may either confirm or refute our conjecture:

Gravity, deg. A.P.I.	33	32	30.1
Flash Point, deg. fahr.	213	200	208
Sulphur, per cent	0.65	0.45	0.84
Viscosity, sec.	42 at 100 deg. fahr.	37 at 100 deg. fahr.	38 at 130 deg. fahr.
Initial Boiling Point, deg. fahr.	434	420	438
End Point, deg. fahr.	714	692	711
Recovery, per cent	98.5	98.5	....
Conradsen Carbon Residue, per cent	0.03	....	—5
Pour Point, deg. fahr.	....	0	0.02

(6) Climatic conditions may be more favorable.

This item may seem to indicate that we are subsidized by the publicity men for which the Coast is noted but, seriously, it seems to us quite probable that there is something in this factor that is of importance.

(7) The need for low costs has caused operators to give closer study to the requirements for successful operation of this type of equipment.

Without making any comparisons, which are not in good taste, it is fair to state that West Coast operations have fostered many of the important developments of truck transport. Whether this condition is due to a more venturesome nature or to the nature of the conditions in which the West Coast operators find themselves is debatable, but the fact is not. By far the larger number of Diesel units operated in this country are in service on the Coast, and we who are interested in the development of this type of equipment are indebted to these pioneers who have pushed this proposition with such determination.

The equipment that is being discussed is of two types:

A six-by-two truck with a six-wheel trailer and a six-by-two tractor with a four-wheel semi-trailer. They are loaded to the gross weights permitted which were indicated previously.

Reports have been received on 22 units that have been in operation for varying periods beginning September, 1935. These jobs have accumulated a total of 750,000 miles in 3,933 car days, an average of 190 miles per day. The largest mileage on an individual unit is 82,000, and the greatest average daily mileage is 377.

Fuel consumption varies from 5 to 8.5 miles per gal. with an average of 6.39 miles per gal. This is such a wide variation that considerable question arises as to the accuracy of the data. An analysis indicates the following:

1 car at 8.5 miles per gal. 1 car at 7.5 miles per gal. 2 cars at 7.0 miles per gal.	4 from 7.0 to 8.5 miles per gal.
4 cars at 6.9 miles per gal. 1 car at 6.8 miles per gal. 1 car at 6.5 miles per gal. 1 car at 6.4 miles per gal. 2 cars at 6.3 miles per gal. 1 car at 6.2 miles per gal. 1 car at 6.1 miles per gal.	11 from 6.1 to 6.9 miles per gal.
3 cars at 5.8 miles per gal. 1 car at 5.6 miles per gal. 1 car at 5.5 miles per gal. 1 car at 5.3 miles per gal. 1 car at 5.0 miles per gal.	7 from 5.0 to 5.8 miles per gal.

From these data it seems fair to accept the 6.39 miles per gal. as a reasonably accurate figure. Unfortunately we have no comparable information on trucks with gasoline engines but, taking this figure as it stands together with the lower cost of the fuel in this territory, very substantial savings in cost of fuels should result.

Reports on mechanical maintenance are meager. Such information as we have been able to gather does not disclose any serious or basic defects.

Some difficulty has been experienced in several of the earlier engines with excessive oil consumption, but this trouble was due to excessive end leakage from the bearings and was corrected readily. Otherwise, the work done is cleaning or repairing injectors and similar accessory apparatus.

The clutch, transmission, driveshafts, and axles have all given results comparable to the service obtained with gasoline trucks, indicating that the Diesel engine of itself does not introduce any peculiar factors as these units were selected for their normal torque capacities.

We have reports on five units operating in the East in which the fuel consumption varies from 5.6 to 8.85 miles per gal. This is also too great a difference to be justified. Three of these units having the same engine are listed as follows:

1 car at 8.85 miles per gal.
1 car at 7.02 miles per gal.
1 car at 5.6 miles per gal.

The other two having another engine:

1 car at 6.7 miles per gal.
1 car at 5.9 miles per gal.

Of these trucks, three are operating satisfactorily. Two have not operated satisfactorily for the reason that mechanical maintenance is so high as to overbalance the savings from fuel. The difficulty has been in the engines, and the engines are not the same. The greatest mileage for any of the five is 100,000 miles and the lowest, 33,000. One engine was rebuilt at 100,000 miles, another at 70,000, this latter being one of the engines that has given trouble. The other troublesome engine has not been rebuilt, but has had ring trouble and fuel-pump and injector trouble.

This very general study of these two groups of cars provides no conclusive information, but does establish some basis for the conclusion that the Diesel engine has been developed to a point where it can be applied successfully to long-distance, heavy-hauling transport.

No information has been obtained on such important questions as to what is the normal expected life up to the time when a major overhaul is needed or whether the equipment will have as long a useful life as gasoline equipment. We know that others have reported on these matters and do not imply that we do not accept their findings, but in this paper we are trying to confine our statements to the experience gained in the units that we have built.

As has been stated, none of these trucks has run over 100,000 miles and, although several engines have been overhauled, this overhaul does not establish the fact that this is the mileage at which overhaul is necessary. Therefore, more experience is needed before this question can be answered definitely.

At this time it will be interesting to continue the analysis of the group of the three cars that were studied in the previous paper. The comparison is not conclusive, but it is instructive. These units are tractors operating in the same service and in the same territory. There is considerable difference in the engine sizes which gives the Diesel units a better performance factor. However, the legal limitations restrict the units to 40,000 lb. gross, and the gasoline unit has sufficient power to do an acceptable job. The drivers on the Diesel units are picked men, having generally better ability than the remainder of the force which is of more than average caliber.

In this discussion, we will use the same designation assigned in the previous paper, namely, Gasoline Unit, Diesel No. 1 and Diesel No. 2.

General specifications are as follows:

Engine	Gasoline Unit Autocar	Diesel No. 1 Diesel	Diesel No. 2 Diesel
Displacement, cu. in.	453	585	672
B.M.E.P., lb. per sq. in.	102.5 at 800 r.p.m.	95.4 at 1200 r.p.m.	108.5 at 800 r.p.m.
Torque, in-lb.	3720 at 800 r.p.m.	4440 at 1200 r.p.m.	5800 at 800 r.p.m.
Transmission	Autocar D5	Brown-Lipe 7351	Autocar TF-5
Gear Ratio, 5th	0.743	0.67	0.75
Gear Ratio, 4th	1.00	1.00	1.00
Gear Ratio, 3rd	1.78	1.73	1.84
Gear Ratio, 2nd	3.52	3.43	3.6
Gear Ratio, 1st	5.85	6.27	5.9
Axle	Autocar CG	Autocar CG	Autocar N
Gear Ratio	7.69	7.69	6.12
Tires	9.75-20	9.75-20	9.75-20
Radiator	Fin and Tube	Fin and Tube	Fin and Tube
Frontal Area, sq. in.	658	658	734
Tubes	210	210	210
Fins per in.	6	5	5
Fan	Six-Blade	Six-Blade	Six-Blade
Fan Diameter, in.	20	22	22
Speed Related to Engine	1.5:1	1.3:1	1.3:1
Tractor Weight, lb.	10,058	11,835	10,799
Gross Train Weight, lb.	40,000	40,000	40,000
Maximum Speed, 5th, m.p.h.	40.0	44.3	49.6
Maximum Speed, 4th, m.p.h.	29.7	29.7	37.2
Gross Tractor Factor, (T.F.), 5th	0.024	0.025	0.029
Gross T.F., 4th	0.034	0.040	0.041

Gross T.F. as used in the preceding table is calculated from the formula:

$$\text{T.F.} = \frac{T \times G.R. \times E}{R.R. \times G.T.W.} \text{ where}$$

T = Engine torque in in-lb.
$G.R.$ = Gear reduction between engine and wheels.
E = Mechanical efficiency taken as 0.90 in direct gear and 0.85 in indirect gear.
$R.R.$ = Rolling radius of tires in in.
$G.T.W.$ = Gross train weight in lb.

A review of these data indicates that the better ability charasteristics of the two Diesel units is due to the greater displacement as the brake mean effective pressure of one is less and the other only slightly higher than the gasoline unit. It is also of significance that the speed at which maximum torque is produced is the same on one Diesel and higher on the other than the gasoline unit. This consideration should allay effectively the idea that better performance can be obtained from the Diesel engine, as such, than can be obtained from an equivalent carburetor engine.

The characteristics as given for the cooling system substantiate the claim of the Diesel for lower losses to the coolant.

The other units such as transmission and rear axle all have performed satisfactorily and do not provide any indication of need for different design factors than are generally used to relate these units to engine size and gross weights.

The transmission on Diesel No. 1 has a torque capacity considerably greater than the ratio of size of the engines and on Diesel No. 2 somewhat less than the ratio of size of the engines, both as compared to the gasoline unit. Both have been satisfactory, indicating that the larger unit used on Diesel No. 1 was not needed.

As for rear axles, Diesel No. 1 has the same axle as the gasoline unit, whereas Diesel No. 2 has a lighter axle. This lighter installation was done advisedly to obtain experience and save weight. The difference in performance indicates that, for satisfactory life, the heavier axle is desirable although, if weight saving is important, the lighter axle can be used.

For the reason that we have been able to obtain more complete data on these three cars we will bring the report that was presented in the previous paper up to date. These data also cover the gasoline unit so that it is possible to discuss comparisons more intelligently.

	Gasoline Unit	Diesel No. 1	Diesel No. 2
Purchase price	100	**125.5, plus 25 per cent**	119.0, plus 19 per cent
Put in service	June, 1934	June, 1935	September, 1935
Total mileage	175,567	79,972	81,591
Maintenance cost per mile	\$0.0160	\$0.0237, plus 48 per cent	\$0.0182, plus 13.5 per cent
Fuel consumption, miles per gal.	3.78	5.96, plus 58 per cent	6.9, plus 83 per cent
Oil consumption, miles per qt.	54.6	62.5, plus 14.5 per cent	59.4, plus 8.8 per cent
Payload capacity	100	92.5, minus 7.5 per cent	96.5, minus 3.5 per cent

These figures are based on the total mileage covered by each vehicle and are quite interesting. Mileage, however, is a very important matter and, therefore, comparisons for approximately similar mileages follow:

	Gasoline Unit	Diesel No. 1	Diesel No. 2
Mileage	80,999	79,972	81,591
Maintenance cost per mile	$0.0122	$0.0237, plus 94 per cent	$0.0182, plus 49 per cent
Fuel consumption, miles per gal.	3.83	5.96, plus 56 per cent	6.9, plus 80 per cent
Oil consumption, miles per qt.	57.8	62.5, plus 8.1 per cent	59.4, plus 2.8 per cent

The largest variation between these two groups of figures is in the item of maintenance. The reason for this variation is because the gasoline unit has had an overhaul at 170,000 miles, whereas the Diesel No. 1 was overhauled at 75,000 miles and the Diesel No. 2 has not yet reached the overhaul period.

We would like to point out the factors that we believe must be taken into consideration in determining the relative merits of such units. Obviously the figures are not complete or accurately comparable, but they can be used for purpose of illustration.

The outstanding item and the one upon which most stress is laid, and we fear too often the only thing considered, is fuel. From the data we see that, for the 80,000-mile period, there is an increase in miles per gallon of 56 to 80 per cent, and these figures are only slightly changed when the Diesel trucks are compared with the total mileage period of the gasoline truck. The cost per mile for fuel on the gasoline truck at $0.13 per gal. is $0.0344 and at $0.10 per gal. for fuel oil is $0.0168 on Diesel No. 1 and $0.0145 on Diesel No. 2. This comparison leaves a saving of $0.0176 per mile on Diesel No. 1 and $0.0199 on Diesel No. 2 and, on a basis of 70,000 miles per year which is approximately the mileage these trucks cover, this saving amounts to $1,232.00 in one case and $1,393.00 in the other.

However, if this fuel is purchased for $0.045 per gal. as we understand it can be on the West Coast, then the cost per mile is reduced to $0.00755 and $0.00652, and the savings increased to $1879.50 and $1951.60 respectively. Furthermore, if the annual mileage is increased to 150,000, which increase is indicated by some of the reports that we have received, then these figures jump to over $4000.00. There is, therefore, some justification for becoming enthusiastic and passing over some of the other factors that may not be so favorable.

The first of these other factors is the increased cost which affects the items of depreciation and interest. Depreciation and how it should be figured is a debatable subject and is treated so differently by various people that it is hard to treat the matter satisfactorily. If the equipment is written off on the same time basis, then the costs during the write-off period will be proportioned in the same ratio as the purchase price. If one piece of equipment has a better useful life after this period than the other, then its costs will benefit. We have not had sufficient experience to support a statement on the matter of comparative useful life. Interest cannot be a sufficiently important factor to influence the comparison for, if this relatively small item represented the difference between the two types, there would certainly be little incentive left in favor of the Diesel truck.

Next is the item of maintenance which appears as $0.016 per mile for the gasoline truck over the full mileage period. This is an increase of 31 per cent over the $0.0122 per mile for the first 80,000-mile period and includes the cost of one overhaul which is also included in the cost for Diesel No. 1. These differences make it hard to provide a true comparison, but it seems fair to assume that the maintenance cost per mile of the Diesel trucks will be nearly 50 per cent higher than the gasoline truck on an equivalent mileage basis. We do not feel that the relatively poorer showing of the Diesel No. 1 is a true picture for either the Diesel engine as a type or of this particular engine. If this assumption is tenable, then there is an increase of $0.006 per mile or $420.00 for the 70,000-mile year or $900.00 for the 150,000-mile year, which must be deducted from the fuel savings.

Oil consumption is relatively unimportant but, from these figures, it is favorable to the Diesel engine. This figure is influenced greatly by the operator's system of draining and changing oil which, if it is done on a time basis, may make considerable difference in consumption on a mileage basis.

The reduction in payload capacity is important because it operates on the foundation of the whole structure – revenue. Unfortunately, we have no precise information on what the gross revenue per mile is in this operation but, if we assume a figure of $0.22 per mile, then Diesel No. 1 loses $0.0165 per mile and Diesel No. 2 loses $0.0077 per mile, which means that $1,155.00 must be deducted from Diesel No. 1 fuel savings and $539.00 from the fuel savings of Diesel No. 2 for a 70,000-mile year and on the basis of Diesel No. 2 would mean a loss of $1,155.00 for a 150,000-mile year.

This item of greater weight and consequent reduction in payload can be mitigated by the use of engines of nearer the same size as the gasoline engine, but this payload increase will be at the expense of the desirable performance characteristics which these larger engines provide. Incidentally, with the Diesel, operators have permitted the use of engines that are really suitable from an ability standpoint, and many operations would be improved definitely if larger and more powerful powerplants were used more generally. Nevertheless, we doubt whether a Diesel truck or tractor can be built successfully with much if any less payload difference than the 3½ per cent figure of Diesel No. 2 and, where there is a gross-weight limit, this increase in equipment weight subtracts directly from the payload and, therefore, from the revenue.

Following we will put these estimates into condensed form so that they can be assimilated more easily, but we wish to reiterate that these figures are not definite and rigidly quantitative but are given to try to illustrate the point that a careful analysis of several factors is needed to arrive at the answer as to whether Diesel trucks are, or can be, more profitable than gasoline trucks.

	Gasoline Unit	Diesel No. 1	Diesel No. 2
Mileage	80,999	79,972	81,591
Fuel cost per mile	$0.0344	$0.0168	$0.0145
Oil cost per mile	$0.00578	$0.00515	$0.00565
Maintenance cost per mile	$0.0122	$0.0237	$0.0182
Depreciation cost per mile	$0.0206	$0.0256	$0.0246
Interest cost per mile	$0.00061	$0.00077	$0.00073
Payload (revenue) cost per mile	....	$0.0165	$0.0077
Total	$0.07359	$0.08852	$0.07138

The fact that these figures are given to illustrate an important point in this discussion, and not as proving that either the Diesel or the gasoline unit is the more economical, can best be shown by pointing out that, if the fuel cost is taken

at $0.045 per gal. or $0.0075 per mile for Diesel No. 1 and $0.0065 per mile for Diesel No. 2, and, instead of a gross-weight limit of 40,000 lb., we take the California limit of 68,000 lb. thereby reducing the importance of the reduction in payload, the whole picture is changed.

Therefore, the point to be emphasized is that the important factors to be considered are (1) fuel cost, (2) maintenance cost, and (3) effect on operating revenue.

We will now revert to the tentative conclusions that appeared in our earlier paper and that were repeated at the beginning of this paper.

(1) That Diesel trucks will weigh more and cost more than gasoline trucks is still true and promises to be so for some time.

(2) The equipment requires different treatment, but the requisite technique is being rapidly acquired.

(3) From the best data obtainable, maintenance is greater for Diesel than for gasoline engines. How much greater it is, as a matter of definite fact, is not yet known. Also, we can only speak for the present and this qualification is important for advance in design and construction is rapid and should act to reduce this difference.

(4) The relative time or mileage to the overhaul period has not been established.

(5) A 50 per cent reduction in fuel consumption is still a fair estimate.

(6) Oil consumption is about equal, but the need for careful attention to draining and to filter elements is important.

(7) The method of driving is important.

(8) The success of changeovers depends on giving careful attention to details.

(9) To these conclusions we should add another, namely, savings from the cost of fuel are certain because consumption is less but against this maintenance and revenue must be checked. Maintenance costs undoubtedly will improve and revenue will be affected least where it is possible to haul heavy loads. Therefore, success is most certain where hauls are long and loads can be heavy.

What has been said probably will be received with dissatisfaction by those who are adherents of either the so-called Diesel cause or the Otto cause. We believe there is no such cause and that, for individuals who are interested in the success of highway transport, the unbiased search for fact cannot have any other result than to further that important, or better indispensable, public service.

The writer wishes to acknowledge gratefully the assistance rendered by his associate, A. J. Scaife, in collecting and analyzing the data contained herein.

Discussion of Bachman Paper

Seeks Dividing Line Between Spheres of Application

– M. C. Horine
Mack Mfg. Corp.

It has been truly said that the more one learns about a given subject, the less he is inclined to make dogmatic statements. Consequently, if Mr. Bachman's conclusions appear to be too much of a "yes and no" character, I think we should reflect that, in view of the character of the man, such an inconclusive summation is undoubtedly an accurate reflection of the character of the subject.

Indeed, as an industrious listener to discussions of Diesel engines on motor trucks for as long as they have had a part in our proceedings, I am bound to state that the definiteness of conclusions presented by other speakers seem to have been in inverse ratio to the amount of information upon which they were based.

Much that Mr. Bachman has told us has been generally accepted as fundamental, namely, that a Diesel engine offers nothing if it does not offer economy, and that such economy can be looked for only in the direction of decreased fuel consumption per mile. Even the most ardent Diesel proponents concede this point.

Where the testimony seems to be confusing is in the question of whether or not Diesel powerplants must inherently be heavier, or more bulky, than gasoline powerplants and whether or not, in the long run, Diesel maintenance costs will be sufficiently greater than those of gasoline powerplants to offset their undoubted fuel economy.

Unquestionably so far, weight, cost, bulk, and maintenance cost of Diesel engines are definitely greater than gasoline powerplants of equal quality and power. Also, undeniably, on long-distance runs carrying or trailing large loads the fuel economy bulks large enough to definitely overbalance these initial handicaps.

The question, therefore, is just where on the scale of daily mileage is the dividing line between the economic spheres of gasoline and Diesel powerplants?

It is all very well to say that the answer to this question will depend upon circumstances but, nevertheless from a practical standpoint, there is unquestionably a fairly narrow twilight zone above which the Diesel must reign supreme, and below which it can have no place.

Personally, I feel that to define this region will require a great deal more definite information than is afforded by the testimony so far at hand. Certainly we will require mileage experience well beyond 250,000 miles, and reliable conclusions cannot be reached on a basis of single installations or comparisons between vehicles operating in different parts of the country.

I believe that Mr. Bachman and Mr. Scaife are to be commended upon the start they have made toward defining the relative spheres of the two types of prime movers in highway transport, and it is to be hoped that they will continue and enlarge the scope of their inquiries so that, by another 12 months, we may have testimony of a much more definite character than their own integrity or the weight of the evidence available at present warrants.

Diesel Truck Exhaust and Taxation Problems Reviewed

– J. F. Winchester
Standard Oil Co. of N. J.

Recently I have had an opportunity to observe a large number of Diesel trucks operating on the highways, and I am inclined to believe that there is a very great need for close study on the part of those who are interested in the fuel characteristics in relation to the ultimate consumption within the unit itself. The need for a good clean exhaust becomes more apparent when we find the general public calling Diesel commercial vehicles the "motor skunks" of the highways. It can be recognized readily that, with such an opinion arising among the large passenger-car highway users, there is going to be an outstanding cry against the use of Diesel trucks unless very close cooperative work is carried on between those who are interested in selling fuel and this type of equipment.

The situation from my viewpoint is very little different than that which I observed some time ago in England. Having studied carefully the growth of the industry at that point, particularly in relation to the taxation problem, I should like to refer you to a complete analysis of what has occurred in the English market since the Crown found it necessary because of reduction in gasoline revenue to increase the taxes on this type of fuel. It is called: "The Influence of Taxation on the Oil Engine," and was published in the Aug. 21, 1936, issue of *The Commercial Motor*. A brief summary of the situation might be of value to the membership at large.

Plant Layout and Production Methods for Modern Aircraft Engines

By J. Carlton Ward, Jr.

Assistant General Manager, Pratt & Whitney Aircraft, Division of United Aircraft Corp.

As will be seen from the title of this paper, its field is extremely broad and, therefore, it will be necessary to divide the paper up into logical sub-divisions, each one devoted to covering only the fundamentals involved. The data presented in this paper are based on the experience of the Pratt & Whitney Aircraft, division of the United Aircraft Corp., at its plant in East Hartford, Conn.

Historical Background

Briefly, this company, which began operations in 1925 in a section of the plant of the old Pratt & Whitney Machine Tool Co., has been a pioneer in the development and production of high-powered engines of the air-cooled radial aircraft type. The first engine developed was one of 400 hp. and was an immediate success, not only because of a material increase in horsepower for an engine of this type, but also because it possessed many unique design features, many of which are retained in those engines now representing the latest designs in this field. By 1929, it became necessary to go into larger quarters and, at that time, the present plant in East Hartford was laid out and constructed.

The original Wasp engine of 400 hp. has since been developed to permit a horsepower rating of 600 hp. The Hornet engine of 500 hp., which was added to the line in 1929, has since been stepped up in rating to 850 hp.

In 1929 the company's engineers foresaw the advantages of two rows of cylinders, and an experimental engine was built and tested immediately. This construction led to a further increase of horsepower totaling 1150 and, at the same time, reduced the overall diameter of the engine.

The more detailed description that follows, therefore, will concern itself with the facilities and methods for production of both single-row and double-row air-cooled radial aircraft engines of the type just described.

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 15, 1936.]

Fig. 1 – Airplane View of Pratt and Whitney Aircraft Factory

THIS paper presents a brief history of the experience of Pratt & Whitney Aircraft, division of United Aircraft Corp., in building high-powered air-cooled radial aircraft engines.

Pictures and diagrams of the plant layout are analyzed in such a way as to show the unusual foresight employed in providing for the future. This arrangement has permitted quick expansion without disturbing production or without revising any of the important features of the original layouts.

Engines are not assembled for stock, and schedules are based on customer requirements with special treatment for Army, Navy and commercial engines. The important part played by inspection and quality control is outlined together with the method of fitting it into the production scheme and the method of meeting the special requirements of Army and Navy.

An appendix includes typical operation sheets illustrating in detail the method of manufacture used to meet the high-performance characteristics required in a master connecting-rod, as illustrating typical problems encountered in the production of an engine of this character.

Plant Layout and Facilities

The Pratt & Whitney Aircraft factory is a unit of a large development consisting of 585 acres of land which includes a modern airport and overhaul station, a large hangar for experimental planes, a manufacturing unit for the production of Chance Vought planes, and a manufacturing unit for the production of the all-metal Hamilton Standard propellers. See Fig. 1.

The factory building for the engine production unit is of a most modern design. It has been thought out carefully and planned as a unit that can be expanded to many times its present size without destroying the fundamental layout or disarranging the flow of material or the relationship among administrative, engineering, research, and production functions. This arrangement is somewhat unique and can be understood better by reference to the schematic floor-plan layout shown in Fig. 2.

The first building is the Administration Building and is entirely an office unit. It can be expanded upward or longitudinally outward without affecting the remainder of the plant. See Figs. 1 and 2.

The Engineering Building is immediately to the rear, separated only by a garage and a passage. It includes patent and research work in addition to all phases of engineering. It also is capable of expansion independently of other sections of the plant.

The factory building is constructed so that a mezzanine approximately 40 ft. wide runs across the front. On this mezzanine will be found all of the shop administrative offices, such as those of Works Manager, General Superintendent, Chief Inspector, Material Manager, Army and Navy, Production Engineering, and all allied functions having to do with the management and control of production.

In Fig. 3 will be seen a typical view of the inside of the main factory building. The present unit is 400 ft. wide and 1000 ft. long spaced in generous bays. The roof construction is such that lighting is uniform and practically independent of the location from the side walls. The floors are of wood block throughout. It also will be noted that there is a generous aisle, and this aisle has been proportioned to take care of practically all of the movement of material as well as of the men when coming on and off shifts. Toilets and washrooms are small mezzanines conveniently located throughout the factory buildings so that they will not form obstructions to the floor plan.

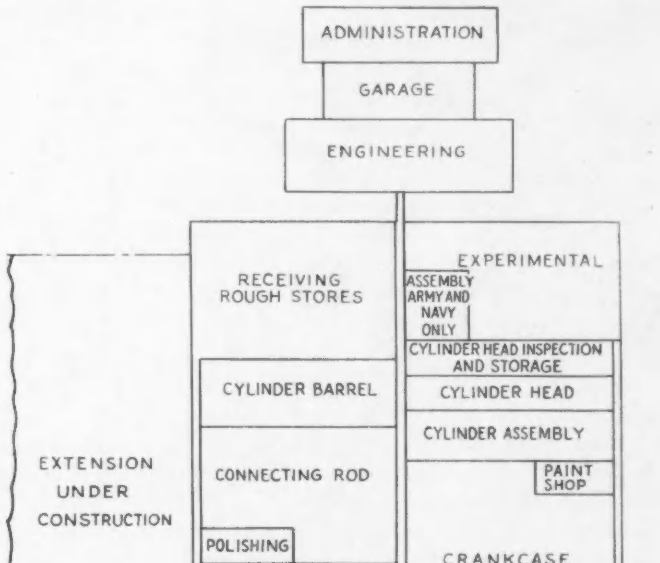


Fig. 2 – Schematic Floor-Plan Layout of Engine Division of Pratt and Whitney Aircraft – Dashed Lines Indicate the Hamilton Standard Propeller Extension Now Under Construction

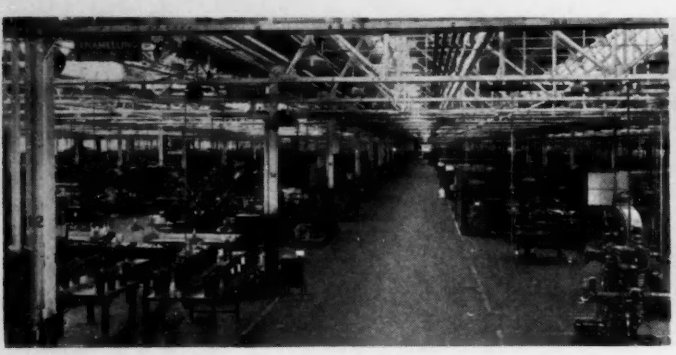


Fig. 3 – Typical View of Inside of Main Factory Building

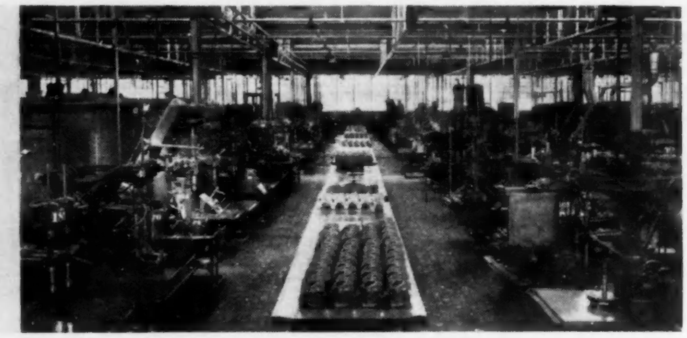


Fig. 4 – Cylinder-Head Machining Line

The plan of the factory is essentially very simple and direct. Under the factory office mezzanine and across the floor on one side is found the Experimental Machine Shop and its assembly shop. They are surrounded by an opaque partition for secrecy. The Experimental Department is a self-contained unit and admission is permitted only to its workers and to certain shop executives and engineers. A separate division of the assembly floor is partitioned off inside the main experimental unit and is devoted only to military and naval engines. See Fig. 2.

On the opposite side of the floor of the shop are found all of the facilities for incoming raw materials including the inspection thereof. Material is scheduled to arrive as needed, and very few stock items are carried in raw-material classifications.

The main manufacturing floor is laid out so that each major component of the engine or allied group of components is made in a separate department which, in most cases, produces the given item or items complete.

There is a minimum of utility departments such as the Metallurgical, Heat-Treating, Painting, Polishing, Toolroom, and Maintenance units.

Cylinder Departments

For example, the first shop department immediately following the raw-material inspection and the Experimental Department is the Cylinder unit. On one side of the main aisle are produced all of the steel cylinder barrels. On the other side are produced all of the aluminum-alloy heads and, in addition, the barrels are assembled to the heads, together with valve seats, bushings, valve guides, and other minor parts so that, when the cylinder is ready to leave the department, it can be final-inspected by the Pratt & Whitney Inspection Department, and by the Army or Navy in addition as the case may be. The completed cylinder assembly then proceeds directly to the Finished Stores Department.

Light alloys of aluminum and of magnesium are machined as far as practical on the north side of the shop, and alloy steels and other hard alloys, on the south side of the shop. This arrangement tends to keep chips and cutting oil solutions conveniently grouped.

The Cylinder Department is followed by the Master and Link-Rod Department, the Crankcase Department, and these in turn by the Crankshaft Department, the Piston Department, the Rockershaft and Valve Guide Department, the Cam Department, the Miscellaneous Department, the Grinding Department, and the Gear Department, as will be noted by a study of Fig 2.

The Tool Room located near the center of the shop is divided up into three functions: first, cutter-grinding; second, tool supplies; and third, tool repair and machine repair. Next to the tool room may be found the electrician, millwright, pipe fitters and other strictly maintenance departments. Since all machines are motor-driven, only a very small belt maintenance section is required.

Between the manufacturing departments and the Assembly Department is the Storeroom layout. This layout is highly functionalized and will be discussed in more detail later under manufacturing methods. All parts flow through the Storeroom to the Assembly Floor or to the Shipping Department as the case may be.

The completed engines are wheeled on their assembly stands through a soundproof wall into the test houses. Between the main manufacturing building and the test houses will be found a unit of the building designed to accommodate a railroad track for car loading and to take care of engine boxes, engines, and storage of like nature.

These test houses running across the rear of the shop are designed to take care of all model engines from the Wasp Jr. to the Twin Wasp, and are constructed in blocks of four with a common control room. Large non-shatterable glass windows permit the test engineer or operator to see the engine while running. The remainder of the walls are of a very heavy masonry construction and are reinforced at the propeller location in such a way that explosion of a propeller will neither endanger the operator nor wreck the house.

Engines are not built for stock but only upon sales specifications and, therefore, no major space need be provided for the storage of completed engines.

The test houses are divided into either production test houses or research and experimental test houses. In connection with the research and experimental test houses, there are a number of dynamometer rooms for engine calibration and detailed engine-performance studies. A large office is provided in this section for test engineers, and again it will be seen that either the production house or the experimental layout can be expanded without interference of one with the other.

Thus the main factory may be described as a single large roofed-over area with uniform lighting and ventilation and with practically no interior partitions, fire walls, and so on, and finally with practically all of the equipment motor-driven. The manufacturing equipment, therefore, can be moved about quickly as older designs of engines are superseded, thus readily conforming to new sequences of operations.

The exterior walls of the building mean very little in the layout and can be pushed out indefinitely as the need for further facilities develops. Each department, in turn, can

expand since the departments run across the shop. Thus, the expansion of each department does not interfere with the adjoining departments.

After nearly six years of use, it still may be said that the proportions of the main building plan are such that, by pushing out the two side walls, production can be built up automatically without destroying the balance between departments or the present departmental layouts.

A review of the general ground plan, Fig. 1, will show readily the available area for extensions. It follows that further expansion can be carried on without the interruption of manufacturing activities in any important degree.

Scheme of Production

As can be seen from the foregoing, under the discussion of plant facilities, the scheme of production is a relatively simple one. Raw material is received by rail or truck through the front of the shop and then flows through the various manufacturing departments to the finished storeroom at the rear.

Conveyors are not practical because of the possibility of damage to the high finish of the parts and because of the fact that many soft alloys are handled. Furthermore, in most cases, production is not high enough to warrant such devices. All parts are handled carefully on electric trucks, almost entirely through the main aisle that runs down the center of the shop.

The storeroom functions are somewhat complicated because of the separate engine and spare-parts requirements both for the Army and Navy and for ordinary commercial requirements. The Army and Navy each maintains a considerable staff of inspectors for whom complete facilities are provided. To meet Government regulations for military or naval engines, all parts must bear the stamp of approval of these inspectors in addition to that of the Pratt & Whitney inspectors before they can be accepted by the storeroom.

Thus a considerable area called a dispatching floor is provided in the storeroom to which all finished parts are delivered whether purchased or manufactured. This dispatching floor serves as a distribution center, and parts are delivered from there to the commercial spare parts and service stores section, the commercial engine parts section, or to the Army and Navy storerooms, called Bond Rooms. Thus, parts are separated functionally immediately upon receipt by the storeroom. Engine parts are segregated one month prior to the assembly schedule in a separate division called the Segregation Storeroom. This division also serves as a ready means for physically checking shortage lists so that, later on as engines are scheduled for assembly, complete sets of parts will have been made available.

In addition, the storeroom is tied in with the packing and shipping of parts. Such parts are inspected and boxed and then delivered to the shipping platform for commercial service or spare parts. Army or Navy spare parts are packed and delivered directly from the Bond Rooms.

It also might be of interest to point out that all the shortage lists and schedule sheets are made on electrically operated Hollerith machines from punch cards made in the storeroom office so that these lists are printed and supplied to the Schedule Department, and Follow-Up Department, in a neatly printed, segregated form and without delay. This feature is a major factor in the efficient control of shop production.

It will be seen readily that the entire set-up is such that, with the exception of certain parts manufactured for spare parts commercial requirements, all material is procured and manufactured against sales orders and in accordance with a carefully predetermined sales schedule.

Purchased raw and finished parts are specified to the vendor as to delivery in separate releases, thus giving an efficient control of inventory. Theoretically therefore, incoming raw material should not rest but, when received, should proceed immediately through the manufacturing operations, the inspection operations, the storeroom operations, and be delivered to the assembly floor or the shipping floor. Finished parts (except for commercial spare or service requirements) also should be received only in time for inspection and delivery to the Commercial Segregation Storeroom or to the Army or Navy Bond Room in time for sales-schedule requirements.

Two important considerations determine this manufacturing policy: The first is that designs are constantly changing, changes being dictated to a large extent by service conditions. The second is that airplane engines are tied in with the airplane design in such a way that each order is custom-built, to a greater or lesser extent, although largely of standard parts.

The question often is asked as to whether costs could be reduced through greater use by the aircraft-engine industry of the straight-line production methods and the quantity production methods used in older and larger industries. The preceding limitation, however, so far has resulted in emphasis on flexibility of operations to meet sudden changes of design rather than on the ability to manufacture quantities of a given type or even of common parts. This policy will be evident from the more detailed comments to follow.

Many Inspections Necessary

The peculiar requirements of an airplane engine are such that an unusual amount of inspection in the form of raw material, finished parts, and finished assemblies is demanded. Thus, each separate manufacturing department has its own inspection group. All inspection groups, in turn, report to the Chief Inspector who is a member of the Engineering Department. No shop or production executive has any control over inspection requirements or any decisions of the Inspection Department. Individuals responsible for engine design are thus made responsible for engine quality.

Each engine upon being assembled completely is run in a test house for a period of from 8 to 13 hr., depending on the type, and is then disassembled completely. The parts are inspected minutely, and the engine reassembled. The engine is again run from 5 to 12 hr. during which it must meet certain performance characteristics and, if successful, it is ready for shipment.

Following this test, the engine is filled with oil in such a way as to prevent internal corrosion. This filling is done on a specially constructed stand with the oil electrically heated to the right consistency. Then the engine is placed in a carefully designed and rugged box, being fastened to a steel plate by the engine mounting bolts. Finally, accessories are put in smaller compartments around the engine and, in the case of export, the boxes are lined with a moisture-proof material.

No castings or forgings are manufactured at the Pratt & Whitney Aircraft shop, and such other parts as are adapted peculiarly to manufacture by specialists are purchased on the outside subject to rigid inspection requirements and supervision.

Development of outside sources is a function of production engineering to insure the control of quality necessary for all

parts entering into the construction of an engine. The Purchasing Department issues orders and arranges all of the financial and accounting details.

It is felt that in this way the greatest efficiency and flexibility are maintained in the manufacturing operations and, at the same time, highly specialized manufacturing operations remain in the control of specialists who are equipped with years of experience and laboratory data for their solution. Thus, quality inspection safeguards are maintained as though the parts were manufactured in the Pratt & Whitney Aircraft shop.

Reference to Fig. 4 showing the arrangement of the cylinder-head machining line will perhaps serve to illustrate the method of parts manufacture employed in the factory. Although the production of engines is in fair quantities, the problem is not one of straight-line production. Thus, for instance, with spare-parts requirements, there are approximately 50 separate active cylinder designs. The equipment has been laid out so that the machines are in sequence and the raw material proceeds in a straight line. Naturally, all machines are not required for any given cylinder. It is felt that this approximation of straight-line production with the many varieties which the factory is required to produce is the best compromise for a problem of this character.

Similar principles are used in all departments with the exception of such departments as Heat-Treating, Painting, Miscellaneous, and so on, which are of a general utility character.

Detailed Problems of Manufacture and Tooling

In endeavoring to illustrate the detailed problems involved in the manufacture of parts and the type of tooling required, practical examples would seem to be more illustrative than a general description. For this reason the master rod of a twin-row engine has been selected as a part that will bring out many of the problems involved in manufacturing this type of engine.

Reference to Fig. 5 will serve to illustrate not only the completed part but the character of finish as well. Many times the question has been raised by practical manufacturers as well as by laymen as to whether the finish is not refined to a greater degree than necessary, and sometimes the question is asked if this high finish is not made to impress the user with the degree of care taken in the manufacture of the parts. Such is not the case however.

Thus an engine of this character is limited severely by many operating requirements. First, it must be of minimum weight and maximum horsepower. Second, it must be absolutely reliable. Third, it must perform at high horsepower output relative to its rating for long periods of time and always with the ability to perform in an emergency at a maximum of horsepower without failure. Fourth, it must be prepared to perform under the preceding conditions for long mileages between overhauls. Fifth, in all cases, parts must be available for assembly into any engine without requiring any hand work that could affect either the dimensions of the part or the character of its finish.

Such an engine will have a dry weight of 1.1 to 1.2 lb. per hp. developed. Automotive engines by comparison have an equivalent weight of from 3 to 5 lb. per hp.

The foregoing requirements preclude any unnecessary strengthening of parts in an airplane engine in order to play safe. The parts must be prepared to stand maximum stresses at all points. These stresses are not continuously applied stresses but are impulsive stresses occurring many times per second. This is an ideal condition for fatigue failure.

Any surface, therefore, that contains tool marks left from milling cutters or other cutting tools, or a condition where one surface blends into another sharply or where edges are not broken properly, is a potential cause for a fatigue failure.

Highly stressed parts when of magnetic material are all magnafluxed. For proper inspection, therefore, surfaces must be of highly finished character in addition to the need for prevention of tool marks leading to fatigue failure.

For those who are not familiar with magnaflux inspection technique, it is sufficient to say that the part must be of magnetic material. The piece in question is then placed between the poles of a powerful magnet in such a way as to set up permanent magnetism within the part. The part is then either sprinkled with fine particles of iron or immersed in a bath containing fine precipitated magnetic iron oxide. Wherever a crack or a seam runs along the surface, even though completely hidden to the eye, the iron particles will gather at this point due to interrupted magnetic flux. Thus, hidden defects become very plain to the eye. Fig. 6 is unretouched and shows a minute seam that proved invisible during machining and during all regular inspection operations, and was not revealed until the part was given a final magnaflux test. It is obvious that this part in a short while would have failed completely and, in addition, ruined the power section of the engine in which it was assembled. All parts of this character are therefore magnafluxed 100 per cent.

Experiments with polarized light on transparent stressed material, in recent years, have told the designer that stresses frequently are concentrated in a manner that escapes mathematical analysis and, to minimize this possibility, it is necessary to blend all surfaces carefully and to avoid sharp corners of all kinds.

The master rod in Fig. 5 is an ideal example to illustrate these points.

It also might be of importance to point out here that modern transport engines are expected to fly 500 hr. or more between overhaul and frequently at an average of 200 m.p.h. or a span of distance of around 100,000 miles.

Coupled with the severity of the operating conditions, this requirement means that parts that have intrinsically a minimum factor of safety must, therefore, be made with a maximum of precision and care.

Reference to Appendix 1 will reveal a summary of operations necessary for the production of the main forging for one of these master rods. Appendix 2 will serve to show a typical operation sheet from which the summary in Appendix 1 has been compiled. The summary of operations in Appendix 1 serves to show the sequence of operations in the manufacture of the main forging of the master rod itself. The sequence of operations for its mating cap and the operations performed after the cap has been assembled to the rod with its proper bolts, and so on, at which time it exists as a machining assembly, are summarized on other sheets.

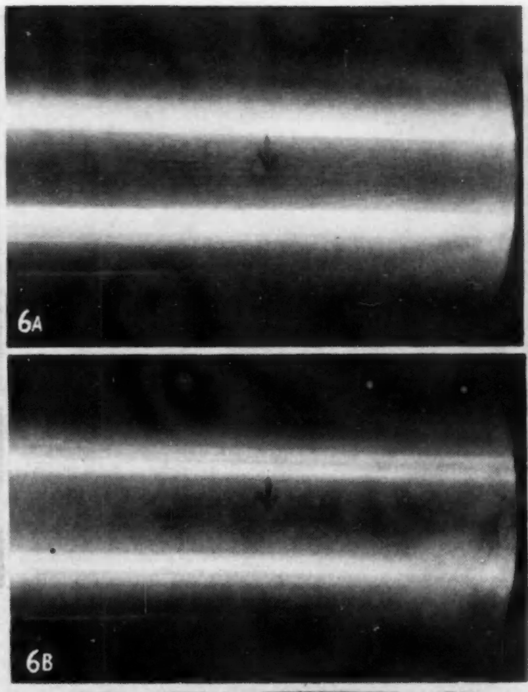
There are approximately 700 different parts to a radial engine of this type, and operation sheets as well as summaries of operations are written in detail for all parts, including their minor assemblies.

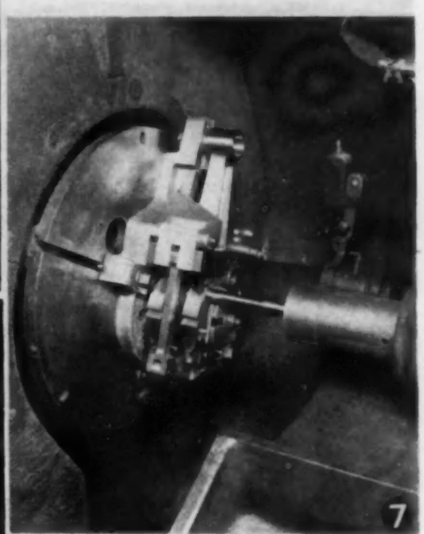
In a paper of this character, space does not permit a detailed discussion of many of the problems that must be met and solved in the production of such parts.

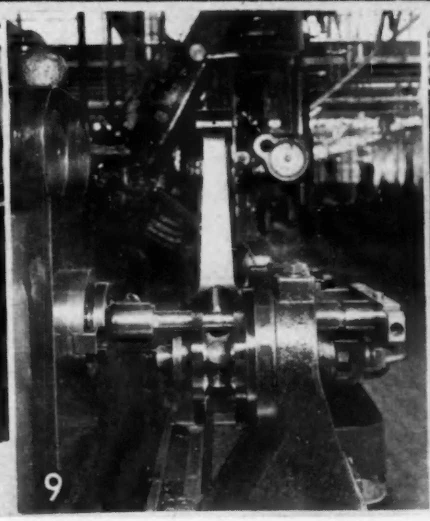
A few general considerations, however, may be of interest: In a forging for this purpose, grain flow is a vital considera-

tion. It is frequently necessary to develop a series of dies before the proper flow of metal is obtained. It is a requisite in a part of this character that the lines of flow in the forging must not be cut by those machining operations that determine the outside finished surface of the forging in all of the highly stressed regions. The flow is determined by the usual metallurgical methods of cross-section deep etching and macroscopical examination. On all important forgings, telltale coupons are required for individual physical examinations. All forgings are inspected 100 per cent in a Brinell machine.

The metallurgical specifications are drawn up by the Metallurgical Department. First, a chemical analysis for the raw material is selected that has been proved by extended experience. Second, it must be an analysis that is not too







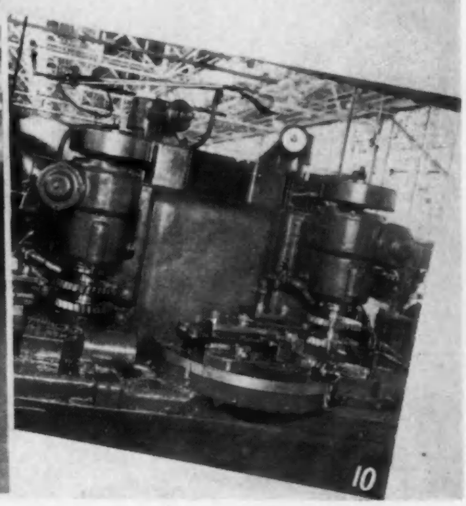


Fig. 5 – Master Rod of a Twin-Row Aircraft Engine Showing the High Finish Required

Fig. 7 – Finish-Grinding the Articulated Rod-Pin Holes on Master Rod

The entire clamping device that holds the cap on the rod floats on four pins so that the rod will not be distorted. In the fixture, holes are located corresponding to the holes to be ground in the master rod. Indexing is accomplished by using these holes in turn on a fixed pin at the true center of the faceplate.

Fig. 8 – Grinding the Parting Line in the Master Rod To Fit Its Cap

Rods and caps are ground in pairs as no tolerance is permitted on this fit. The cap is used as a gage in grinding the rod. The clamping device operates to hold the rod firmly against a half plug representing the crankpin.

Fig. 6 – Wrist-Pin before Magnafluxing (*above*) and after Magnafluxing (*below*), Revealing a Normally Invisible Seam

Fig. 9 – Method of Machining the Master Rod Between the Flanges for Clearance Around the Head of the Articulated Rod

Immediately below the arbor will be noted the effect of a preliminary operation of countersinking the formed counterbore with a drill press. The flycutter shown is used as a roughing tool and is followed by a formed flycutter that is fed first in one direction and then reversed in order to blend the clearance groove with the flanges. The reamed holes in the rod flanges are the locating points for indexing. The outboard bushing for the arbor fits into the reamed holes in the rod.

Fig. 10 – Production Machine Designed Specially for Milling the Ends of All Types of Articulated Rods

Loading stations are provided by means of a turntable. The twin fixtures can be changed quickly to accommodate all the various designs of articulated rods.

sensitive to variations in physical characteristics. Third, it must be a material readily procured in this country in the open market. Thus, the master rod illustrated in Fig. 5 is SAE X-3140 steel.

Immediately after the forging is produced and passed for inspection, it must be subjected to a complete normalizing treatment designed to prevent slow distortion of the material during machining. Where the material is not normalized completely, there have been cases in which it has passed through machining satisfactorily but, when subjected to the loads in its engine, hidden stresses were released causing the part to spring out of shape.

The rough weight of a typical rod forging of this character is 22 lb. and the finished weight is 13.5 lb., indicating the extent to which metal is removed by the machining processes.

Offhand, it might appear that an unnecessary amount of machining is required, until it is appreciated that an effort is made by the designer to provide cross-sections throughout that keep the stresses as nearly as possible to a uniform intensity. It has been found that, where the stress intensity changes abruptly, secondary stresses are induced which have led to failures.

It also may be interesting to know that the complete master-rod machining assembly must be held within a weight tolerance of 0.03 lb. An examination of this permissible weight variation will show clearly that, if a rod were made to maximum dimensions, permissible within the rod tolerances, it would be overweight in regard to the weight specification. Likewise, if made to minimum dimensions permissible within drawing tolerances, it would be below the minimum permissible weight for the rod assembly. A balancing hole is provided from which metal may be removed when the rod is being brought to the specified weight.

Army and Navy requirements demand that parts be marked for identification. In general it is not permissible to use metal stamps as they would cause high stress concentration and bring on fatigue failure. Etching or some similar process must be used in such cases.

On certain parts where stamps are permissible, the location of the stamp is always specified on the drawing with the preceding thought in mind.

Although individual tolerances for a part such as the rod in question are sometimes met in regular automotive practice, nevertheless, there are practically no cases in automotive practice where these fine tolerances are applied to so many dimensions on the same part.

Although it is customary in many industries to work to tolerances specified in tenths of thousandths of an inch, it is seldom that such dimensions are required on more than one important surface on a given part. However, in a rod such as we are considering, there are many dimensions, all of which must be worked to tolerances measured in tenths of thousandths of inches. It is obvious that this is a far more difficult task and is one reason for the complete normalization that is required of the material itself before being worked.

Again it should be brought out that, in addition to the close requirements for size, the finish must be practically perfect. For instance, the articulated rod pin hole carries the entire stress transmitted from one cylinder to the crankshaft. Every portion of the surface of the hole and of the mating pin must be a bearing surface and thus the depth of machining scratches is important.

In the production of this rod it has been necessary to grind all of the articulated rod pin holes in order to produce smooth enough surfaces to meet these requirements. Both the alignment and the diametral variation are of the greatest importance, and each is held to tenths of thousandths of an inch.

A further complication is introduced by the fact that holes in one flange are slightly larger than the holes in the other flange. This variation has been made necessary by the fact that the knuckle pins are an extremely close fit in the master-rod hole and would tend to mar the surfaces during the assembly operation if both holes were the same size.

Fig. 7 illustrates the set-up for grinding these holes with the master plate in the chucking grinder. The fixture is made of aluminum to eliminate centrifugal unbalance.

Another extremely interesting fixture is that for producing the parting line between the cap and the rod. In this case, the fit must be so close that the cap is ground and then serves as the gage for grinding the rod. No tolerances are permitted between the mating surfaces. Each cap is tested for the bearing surface with its mating rod with Prussian blue. A photograph of the set-up for the machine for grinding the rods to fit the caps will be found in Fig. 8.

It also is interesting to note that the connecting-rod bolts that hold the cap to the rod must be assembled without any eccentric loading in view of the high stresses involved. These bolts, therefore, are ground in the threads and on the body on their own centers to insure concentricity between thread and body.

Another unusual operation is that of producing the clearance grooves machined in the master rod to allow clearance around the knuckle-pin end of the articulated rod. A formed counterbore is sunk to the proper depth after which a flycutter is used through the articulated rod pin holes in a suitable arbor which, in turn, is fed in both directions so as to fly-cut the fillet where this surface blends into the flanges.

These operations are illustrated in Fig. 9.

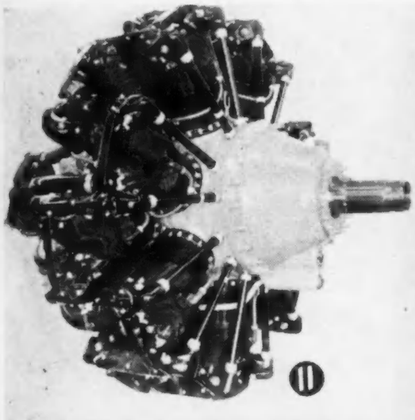
Fig. 10 illustrates a production machine designed specially for milling the ends of all types of articulated rods.

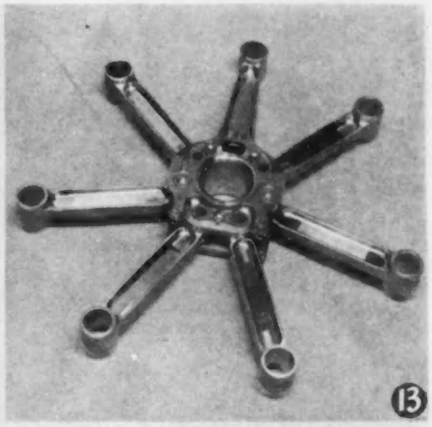
This operation is included at this point to illustrate the type of special machine and tooling that can be used only where the volume of parts permits. In such a case, it is necessary that the machine and set-up be flexible enough to accommodate all of the different designs of such parts. It must also be conceived with the thought in mind that, as the design of the part is modified, the machine and set-up can be modified readily to suit. This operation is, therefore, more in the nature of a quantity-production operation than the others illustrated previously.

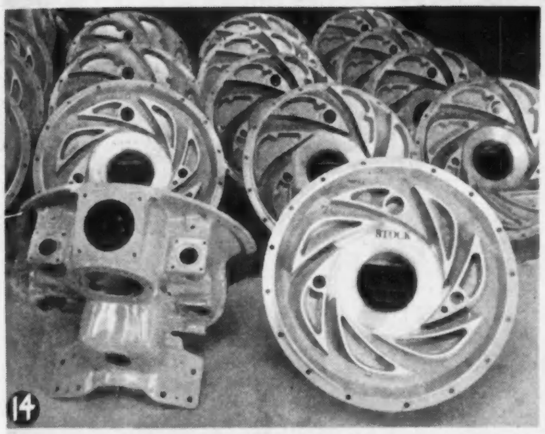
It should be evident at this point that the first manufacturing consideration is the production of a precision part in order to meet the severe requirements laid out by the designing engineers who have been influenced in turn by the exacting performance characteristics required of the finished engines.

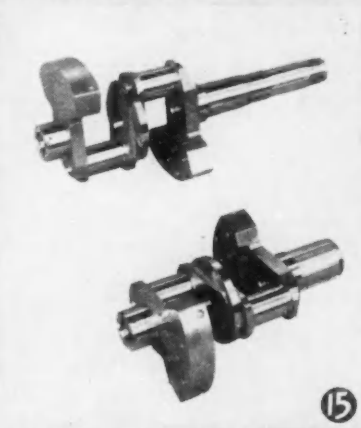
However, when it is possible to work in a production fixture or special machine, advantage is taken of such an opportunity. As has been stated elsewhere in the paper, this latter condition is more of an exception than a rule so that, as a natural result, the bulk of the manufacturing equipment is standard general-purpose machinery that can be adapted rapidly to widely different designs of parts.

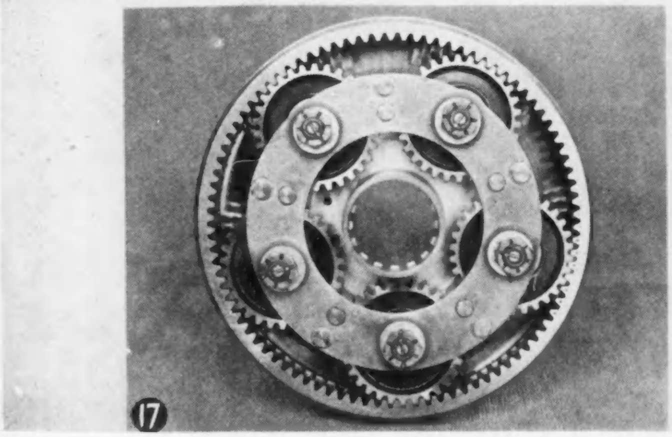
Although the twin-row engine master rod has been used in the foregoing to illustrate typical manufacturing problems involved in producing the type of engine under consideration, it also may serve to give a broader understanding of the problem by selecting for illustration, other parts as well. Space does not permit further detailed discussion and in order, therefore, to give a wider view of the problem as a whole, additional photographs of typical parts, tools, and manufacturing operations are presented in the illustrations that follow.











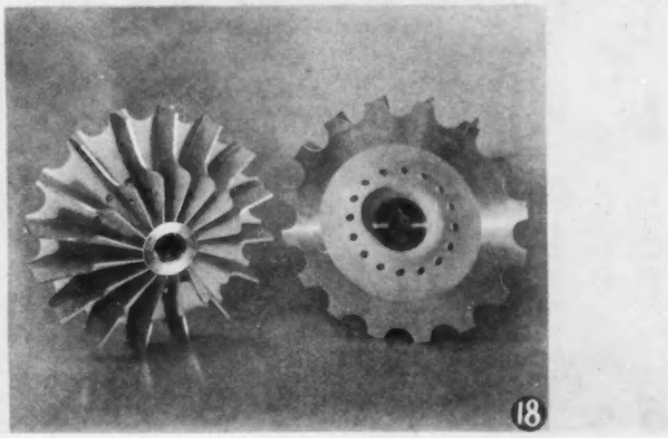


Fig. 11 – Pratt & Whitney Twin Wasp Engine

Fig. 12 – Crankcase of Aluminum-Alloy Forging Requiring a Number of Difficult Machining Operations – Steel Bearing Liners Are Shown Ground in Place

Fig. 13 – Rod Assembly – Method of Assembling the Articulated Rods into the Master Rod and Cap

Fig. 14 – Rear Section of Crankcase

This section may be either an aluminum-alloy or a magnesium-alloy casting. Wall thickness must be held more closely than with ordinary foundry practice. No adjustable parts for assembly purposes or selective assembly are permitted in assembling the parts into the rear section. All gear distances must be produced accurately in the original machining. Thus all rear units are completely interchangeable. Note the steel deflector vanes cast in place in the carburetor-intake elbow.

Fig. 15 – Crankshafts – (*above*) Direct-Drive Type; (*below*) Gear-Drive Type

These crankshafts are normalized alloy steel and, when assembled with their counterweights, are dynamically balanced and weighted for their master-rod and piston assembly.

Fig. 16 – Cylinder Assembly

The aluminum-alloy heads are hot-shrunk on steel barrels and, while hot, the valve seats, valve guides, rocker-shaft bushings, and so on, are inserted. The head casting is made with 5 fins per in. Delicate fin cores are required with as much as 5 in. draw, making this a most difficult casting to produce.

Fig. 17 – Cage and Pinions

This assembly must transmit over 1000 hp., and is an example of extreme precision in manufacture. All important gears have ground teeth throughout the engine. The bell gear is an example of internal tooth grinding.

Fig. 18 – Impeller

This impeller rotates at a tip speed approaching the speed of sound and must be dynamically balanced with extreme care. It is driven by precision gearing with ground teeth.

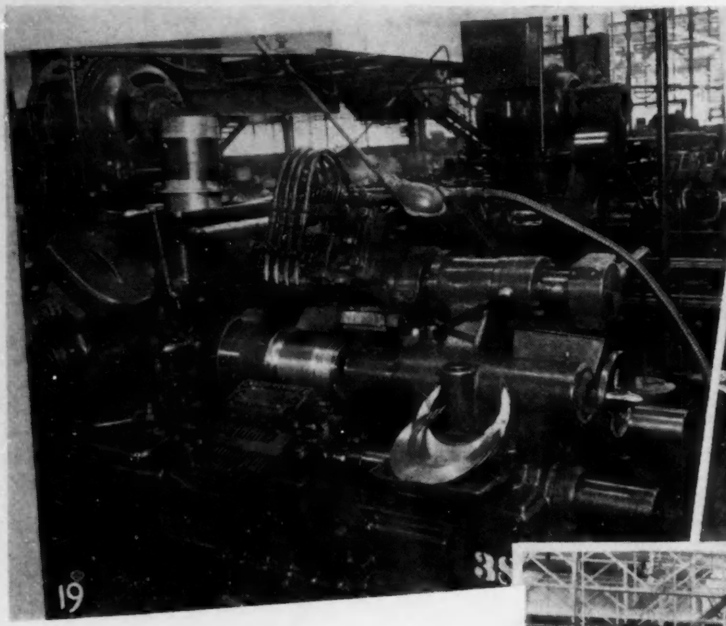


Fig. 19 – Auto Lathe for Turning Cooling Fins on Cylinder Barrels

Fins when completed are 0.020 in. thick and, as will be noted, are turned from the solid forging. After rough-turning, the forging weighs 33 lb. and, when fully machined, it weighs only 7.8 lb.

Fig. 20 – Special Paint Oven

This electrically heated conveyor-type oven is designed to dry the various primers and paints used in connection with resistance to salt-water corrosion and to meet Army and Navy requirements. Careful attention must be given to cleaning before painting. Dichromate etching is required for magnesium-alloy castings prior to the painting operation. The Paint Department is shielded with glass partitions to eliminate the effect of dust particles carried through the air from the remainder of the shop.

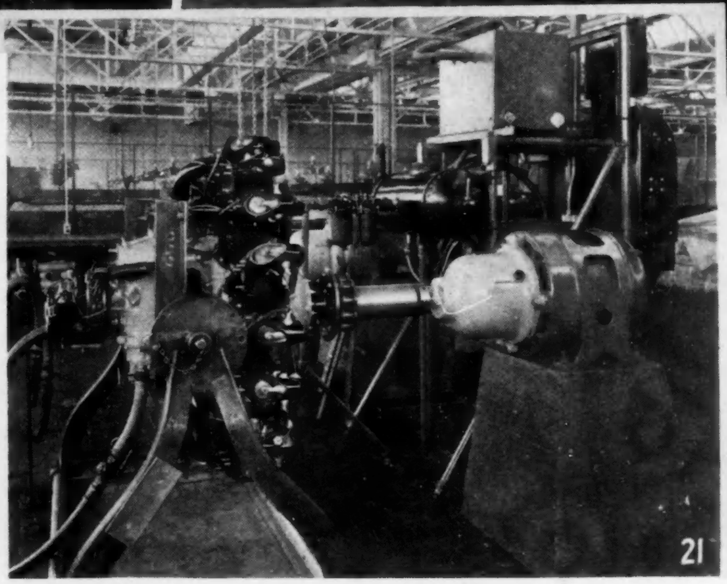


Fig. 21 – Special Running-In Stand for Cold-Running the Assembled Engines

The photograph illustrates the first running-in test given to the assembled engine. This operation tests the functioning of the moving parts and, due to the low clearances in the engine, assures proper oil circulation before the engine is fired.

With this thought in mind, three separate groups of illustrations are included herewith.

(1) Illustrations of the engine or typical parts of the engine, Figs. 11 to 18 inclusive.

(2) Illustrations of certain of the more special machines with their tooling, Figs. 19 to 21 inclusive.

(3) Illustrations of typical fixtures and tools, Figs. 22 to 25 inclusive.

Thus, the rods, crankshaft, cylinder barrel, and reduction gearing are typical examples of alloy steel parts with intricate machining operations.

The crankcase is worthy of study as an example of one of the most difficult as well as one of the largest aluminum-alloy forgings produced in this country.

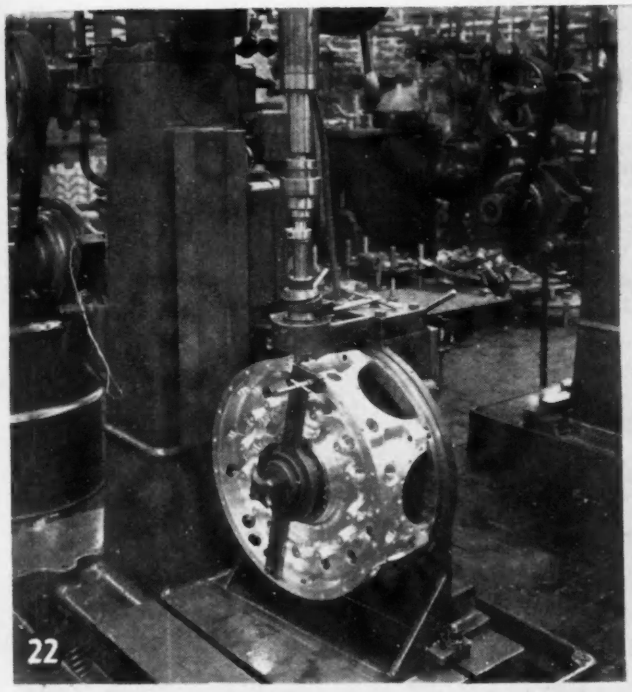
The crankcase rear section has been included (Fig. 14) to illustrate a typical magnesium and aluminum-alloy casting. The machining of such light alloys is a typical specialized problem entering into the production of this type of an engine. It is possible to have dangerous fires among the chips from magnesium-alloy machining operations, although this possibility has now been reduced to a very minor hazard by taking proper precautions.

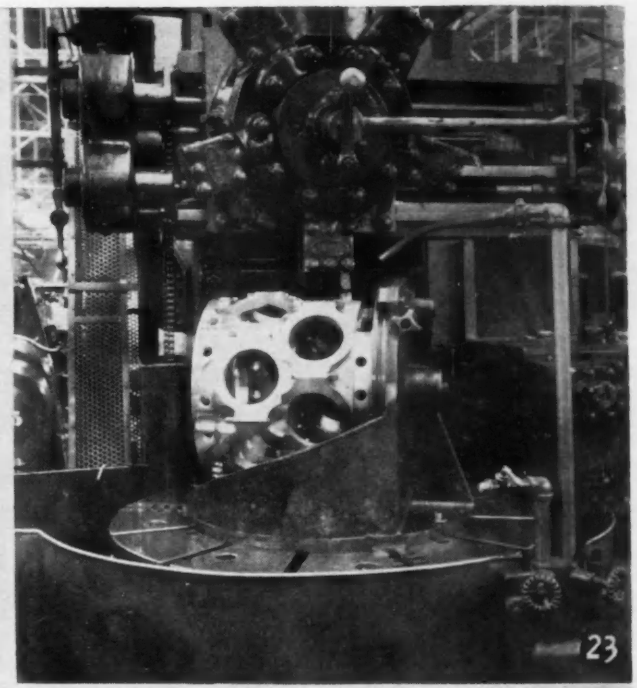
Due to the fact that, with temperature variations between summer and winter in the machine shop, magnesium alloys tend to expand and contract at a greater rate relative to steel and iron, a serious problem is introduced in holding the required accuracy in the overall operating temperatures. It has been found necessary to shut down on machining operations affecting magnesium alloys when the shop temperatures exceed a well-known definite critical point.

Such castings as the rear section must be gaged thoroughly, not only during the process of core-setting in the foundry but before passing finished castings on to their machining operations. Porosity tests are applied, and wall thickness must be held to limits much less than those met with in normal foundry practice. Special inspection gages are provided by which these wall thicknesses can be measured on the rough castings.

In the second group of illustrations, Figs. 19 to 21, will be seen certain machine tools or manufacturing equipment that have been adapted specially to the manufacture of parts of radial aircraft air-cooled engines.

An attempt is made in each photograph to show the machine, its tooling, and the part being machined in order to





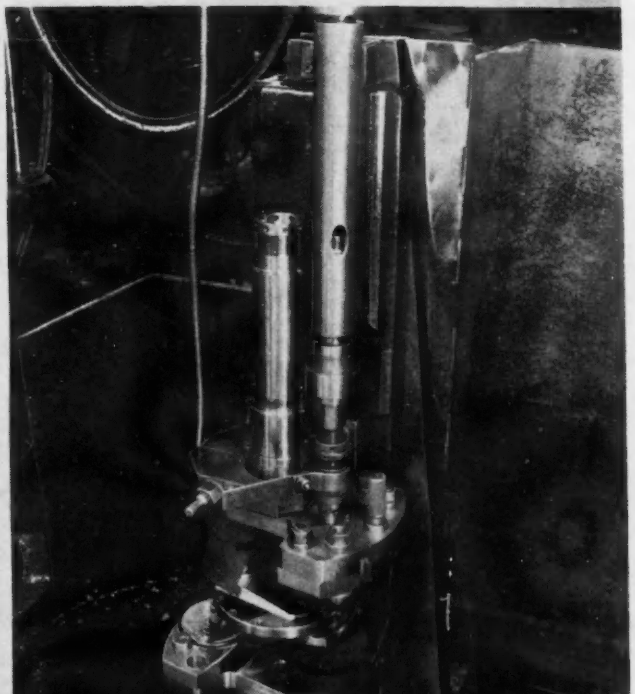


Fig. 22 – Special Drilling Fixture for Tappet Guide Holes in the Crankcase

The angular position of these holes must be held within a cumulative tolerance of 0.003 in. The work is located diametrically on the flat index plate by means of a central plug fitting the main bearing liner, and angularly by means of a plug fitting through the front strap into a dowel hole in the face of the case. This fixture is used for both front and rear sections of two-row crankcases. The rear section is shown.

Fig. 23 – Fixture for Boring and Facing Cylinder-Barrel Holes in Main Crankcase on Vertical Turret Lathe

This fixture has been adapted to five sizes of crankcases. The fixture is located in the center of the table by a pilot that fits the hole in the table. The work is located on the flat index plate by the main bearing liner and one dowel, and is held securely by several clamps fitting into the valve tappet holes, and by a large strap in the mid-section of the crankcase. Indexing is provided by a spring plunger housed in the rear casting of the fixture.

Fig. 24 – Push-Rod Hole Boring, Facing, and Tapping Fixture

Nine fixtures are designed flexibly to produce 50 different cylinder castings. This is a swinging-type index fixture. Location of the work for center distance is from the outside diameter of the lower flange which is held within ± 0.001 in. for this purpose. Plugs placed in the valve guide holes and fitting into a central slot in the back of the fixture provide positive angular location. The work is held rigidly against the back of the fixture by the substantial hinged strap shown in front. Note the universal joint provided for the boring bar to obviate the possibility of cramping the bar.

Fig. 25 – Drilling, Reaming, and Countersinking Fixture for Counterweight Rivet Holes for All Models of Twin-Row Engine Crankshafts

This is a large swinging-type fixture in which one cheek is machined and the face of the fixture with the work swings through 180 deg. for machining the other cheek. Both the crankshaft cheeks and the counterweights are rough-drilled before this operation, and the counterweights are bolted to the cheeks through four of the holes. The remaining holes are finish-drilled, reamed, and countersunk, and the bolts transferred to these finished holes for completion of the operation.

Appendix 1

SUMMARY OF OPERATIONS

PART NAME Master Rod PART NO. 15232 1830-B

MATERIAL P. W. A. #191 Steel Forging 15232-F DATE 8/30/35

TYPE OF MACH.	DESCRIPTION OF OPERATION	DATE CHANGED	OPER. NO.	DEPT.	TIME	RATE	UNIT	
	Operation 1 is rough forging inspection							
Surface Grind	Surface grind sides		2	12 or 6				
Sund. Miller	Straddle mill wrist pin end		3	14				
Drill Press	Drill, re-drill and ream wrist pin hole		4	14				
Drill Press	Rough bore 2.906 & Finish bore 2.886 dia's.		5	14				
Plain Miller	Mill cap seats		6	14				
Barnes Dr. Pr.	Bore $1\frac{1}{2}$ and 1" radii		7	14				
Barnes Dr. Pr.	Rough and finish 1-3/4 rad.		8	14				
Barnes Dr. Pr.	Rough drill and C'bore in "I" section		9	14				
Vert. Miller	Mill top of web @ 2° 33' 46" (both sides)		10	14				
Vert. Miller	Mill both sides to blend with $1\frac{1}{2}$ and 1" radii		11	14				
Vert. Miller	Rough mill "I" section, both sides		12	14				
Vert. Miller	Finish mill "I" section, both sides		13	14				
Barnes. Dr. Pr.	Spot & Dr. knuckle pin holes, Dr. & R. lightening hole		14	14				
Barnes. Dr. Pr.	C'bore knuckle pin holes both sides		15	14				
Vert. Miller	End mill 18° 35' ang. to blend with 1-3/4 rad. (both sides)		16	14				
Drill Press	Drill and ream. 680 rad. both sides		17	14				
Vert. Miller	Rough circular mill between flanges		18	14				
Vert. Miller	Rough circular end mill 3-19/64 rad.		19	14				
Vert. Miller	Finish circular end mill 3-19/64 rad.		20	14				
Plain Miller	Straddle mill cap seats		21	14				
Plain Miller	Straddle mill cap seat chamfers		22	14				
Vert. Miller	Circular mill hub contour, both sides		23	14				
Vert. Miller	Rough mill wrist pin end		24	14				
Vert. Miller	Finish mill wrist pin end		25	14				
Vert. Miller	Mill off surplus stock from wrist pin end, both sides		26	14				
Drill Press	Form 1/32 rad. on top of wrist pin boss		27	14				
Avey Dr. Press	Dr. (1) #19 (.166)	12/13/35	28	14				
Avey Dr. Press	Drill (2) 5/16 holes (wrist pin end)	"	28-A	14				
Electric Oven	Bake at 550 to 600° F. for $1\frac{1}{2}$ hours		29	16				
	To be Completed Under Number 16800							

give an idea of the set-up involved. In general, the machines in these illustrations show the more specialized type of machines and equipment used. Such a machine as the automatic lathe, which has been fitted up specially, is an example of adapting standard machines to specialized operations.

It should be made clear, however, that highly specialized machines often cannot be provided due to limitations of quantity of like parts to be produced and because of frequent changes in design of the parts. Therefore, the policy is to adapt efficient, modern standard general-purpose machines with tooling designed to produce the given accuracy, and which adaptation allows flexibility for design changes.

The last group of illustrations are those included in Figs. 22 to 25, illustrating certain of the typical tools for some of the important machining operations. It is difficult by means of illustrations to point out the operating features of these tools and space does not permit detailed discussion. However, tool design is made difficult by the fact that provision always must be kept in mind for changes in design of the product and by the need for adapting the more expensive tools to similar parts of different models of engines.

In connection with the machining of the soft alloys, the tools must be designed with great care to avoid marring or springing the work. The design of the tools also must permit the gaging of the work during machining.

Although no detailed discussion can be given here in regard to gages and inspection equipment, it should be obvious that this is another very important consideration. All close limits are provided with inspection and master gages. Complete flush-pin and contour gages as well as the more standard types of optical gages must be provided. These are provided for when the fixtures are designed. Functional gages are included wherever possible.

It follows that, in view of the fine limits and the machining tolerances, it is necessary that the gages be correspondingly precise.

The maintenance of cutting tools is an equally important factor. Cutter grinding, therefore, is a separate section of the tool room and includes a wide diversity of types and sizes of cutter-grinding equipment. Heavy cutter-grinding equipment is required as many of the milling cutters are ground to limits of tenths of thousandths of an inch.

It is hoped that sufficient information has been presented to indicate the need for special accuracy and care in the manufacture of parts for aircraft engines entering into military and transport fields, which accuracy and care are perhaps peculiar to such uses and, therefore, not required in automotive powerplants for other purposes.

It should be obvious that the total demand for engines of this character is as yet sufficiently limited so that the problem is one of producing a mechanism interchangeable in a true sense and yet a mechanism that cannot be manufactured by the mass-production methods which generally have been associated with these requirements.

Appendix 2

OPERATION SHEET

PART No. 15232 PART NAME Master Rod PART No. 15232

MATERIAL P.W.A. #191 Steel Forging OPER No. 9 DEPT. No. 14 No. SHEETS

TYPE OF MACHINE Barnes Drill Press

DESCRIPTION OF OPERATION	NAME OF TOOL	TOOL No.	SKETCH OR REMARKS
Clean locating surfaces, lower support pin, place rod in fixture	Base Plate	15232-T-9	Added "PWA #191 Steel Forging" 9/10/36
and clamp large end lightly	C'boring Fixture	Tam-11	
Locate with stud in wrist pin hole and clamp tight against big stud	1-3/8 dia. Drill	15232-T-10	
Bring support pin up to hub and tighten	1-3/8 dia. Drill Holder	1255-T-18	
Clamp large and tight with "C" washer and nut	1.552 dia. C'bore	15232-T-11	
Rough drill 25/32 rad into "I" section to 1-3/8 dia. and to 3/32	1.562 dia. C'bore Holder	1255-T-18	
high from center of rod to point of drill and holding 4-5/32 center	15/16 dia. Drill & Holder	1255-T-66	
dim.	.990 dia. C'bore & Holder	1255-T-19	
C'bore 25/32 rad to $1.552^{+.000}_{-.010}$ dia. & to $1/16^{+.005}_{-.000}$ high from			
center of rod holding 3/16 rad & 4-5/32 center distance			
Rough drill 1/2 rad to 15/16 dia. & to 3/32 high from center of rod	1.552-1.542 Plug Gage	Std	
to point of drill holding 1-25/64 center to center dim.	.990-.980 Plug Gage	"	
C'bore 1/2 rad to $.990^{+.000}_{-.010}$ dia. & to $1/16^{+.005}_{-.000}$ high from center	2" Micro. with extension for		
of rod holding 3/16 rad and 1-25/64 center to center dim.	$1/8^{+.010}_{-.000}$ thickness	Std	$1.552^{+.000}_{-.010}$ was 1.562 ±.005 – $.990^{+.000}_{-.010}$ was 1.000 ±.005 – 1.552 was 1.562 – .990 was 1.000 – 11/26/35 1.552-1.542 was 1.567-1.557-.990 .980 was 1.005-.995-1/2/36
Turn rod upside down and repeat above operations			
Note: Thickness of "I" section must be $1/8^{+.010}_{-.000}$ to allow for polishing, all cuts must be smooth			

The Airship and Its Place in Modern Transportation

By Dr. Hugo Eckener
Chairman of the Board, German Zeppelin Transport Co.

THIS paper discusses the position of the airship as a means of transportation, briefly reviewing the history of the supposed but really non-existing competition between the airship and the airplane.

The results of the *Hindenburg's* ten North Atlantic demonstration trips of 1936 are reported on, and the meteorological observations made are discussed. Figures of cost of operation and revenues of the *Hindenburg* are revealed, showing the relatively low cost of operation of the modern passenger airship.

The relative places of the express steamer, the airship and the airplane in the future North Atlantic transport picture are discussed, and the necessity for cooperation between Germany and the United States in the future development of the airship is emphasized.

Since Dr. Eckener presented this paper at the 1937 Annual Meeting of the Society, an increased 1937 schedule of Transatlantic service for the *Hindenburg* has been announced. Eighteen round trips across the North Atlantic, eight more than last year, are planned by the American Zeppelin Transport Corp., acting as U.S. agent for Deutsche Zeppelin Reederei, the operating company.

The Naval Air Station at Lakehurst, N. J., will remain the American terminal.

TIMES have certainly changed and, naturally, I consider that they have improved. For, had I even as recently as two years ago mentioned the word air-transportation, I venture to think that at least 90 per cent of my hearers would have thought only of the airplane and airplane transportation whereas, of the remaining 10 per cent who might have included the airship in their thoughts, again nine-tenths would probably have looked upon the airship as a rather amusing, passing phenomenon which in the near future would have to abandon the field of competition to the airplane.

We in Friedrichshafen always have been of the opinion that the wide oceans between the continents are beyond the "performance ability" for economical airplane transport and, therefore, we decided to construct airships for operation across the oceans. The way we set about our task methodically and step by step is too well known for me to dwell upon it, except to stress the point that in those first years we looked upon our trips as study trips for the purpose of acquiring knowledge and experience in the airworthiness of the airship under all possible meteorological conditions – for the open ocean, with its storms and squalls, was still a terra incognita for all aircraft.

Who at that time knew anything definite as to the forces or potency of ocean storms at flying altitudes or the turbulence of atmospheric conditions in the windshifts and line squalls of the North Atlantic? Who could, with any certainty, foretell how the airship would behave in a hurricane, in the deluge of a tropical cloud burst, or in the thunderstorms of the Gulf Stream regions? These questions had to be solved before we could assume the responsibility for declaring the Zeppelin a safe and reliable means of commercial transport across the ocean, and so we traveled for three years over the seas in all regions and latitudes. The results showed that the airship could always hold its own.

I think it may, perhaps, interest you at this juncture to hear some of our experiences during the ten demonstration trips that we carried out last summer over the North Atlantic because an air service between the United States and Europe cannot fail to be of tremendous importance to you in connection with your foreign trade relations.

The North Atlantic is commonly known as a particularly unpleasant weather area and, therefore, is held to be little suited for regular air service. Two years ago, when I had the honor to be received by your President, Mr. Roosevelt, and I discussed my plan for a series of study or demonstration flights, he told me that, in his opinion as an experienced sailor who was keenly aware of the treachery of the North Atlantic, the establishment of a regular service would be very difficult.

This opinion of President Roosevelt was shared by many people, and particularly by meteorological specialists who had expressed their doubts frequently; and indeed, they were quite right! For the area southeast of the Banks of Newfoundland, at the point where the cold waters and air masses from the Labrador stream meet and merge with the warm waters and air masses of the Gulf Stream, is one of the very worst bad-

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1937.]

weather factories of the world. The low-pressure areas coming from the west generally acquire here unusual intensity, and the storms increase as they move further eastward. They not only develop enormous force, but are accompanied by violent squalls, thunderstorms, and windshifts.

We were thoroughly alive to this situation when we formed our plans to explore this region by actual flights, and we calculated with an average of about 65 hr. for the westward voyage from coast to coast, and 45 hr. for the eastbound trip. Our published schedules were based on these calculated running times.

We experienced a pleasant surprise, for we found that we could average 65 hr. for the entire trip from Frankfurt to Lakehurst – not just from coast to coast – and, when eastward bound, we averaged 50 hr., or 45 hr. from coast to coast.

A very interesting discovery was made, namely, that one was not always obliged to meet with continuous adverse westerly winds, particularly if a course running about 300 miles north of the steamship route – approximately on the fiftieth parallel – was chosen. On this route, which crosses a meteorological region which is little explored as it is traversed rarely and weather reports are but meager, one frequently remains north of low-pressure formations and, therefore, meets northerly and oftentimes easterly winds. In this area over the cold waters of the Labrador stream, line squalls and thunderstorms assume a much milder form, and this condition offers steadier and more pleasant conditions for the flight of the airship. I am, however, prepared to admit that our demonstration trips were summer trips, executed between the beginning of May and the middle of October. Whether it would be possible in the late fall, or even in the winter, to encounter similarly favorable conditions, I will leave in abeyance for the moment. Further study trips in the near future will decide this question.

Why then, one may inquire, if the present attitude toward the airship has become more favorable, do people still hesitate to draw the obvious conclusions from these facts? The answer to this question is, without any doubt, the one which I referred to earlier in the paper, namely, because many critics of the airship contend that in the very near future the airplane will be able to take over the mail, and probably also the passenger service, across the North Atlantic. In the opinion of these critics it would, therefore, be useless to construct expensive airships and airship terminals which would be rendered obsolete in a short period of time.

Far be it from me to disclaim the possibilities of further airplane development. On the contrary, I will be happy with each step forward, enabling improvement of service. But I must confess that I am unable to recognize at present a solid basis for the propounded hopes in this direction. On the other hand, the airship, which is now able to render trans-oceanic service, is already here, and it is only necessary to take hold! In my opinion, one should hesitate the less since *it is not at all certain that the airship will have finished its usefulness at the moment at which the airplane will also be in a position to transport a commercial load of passengers, mail, and freight across the oceans.*

This is certainly today the decisive question on which the future of airships depends, and it is, therefore, necessary to go more deeply into the matter. Since the airship has relinquished voluntarily all competition in short and medium flights to the airplane and has confined itself to long non-stop trans-oceanic trips – a field in which the airplane cannot now offer safe and regular service – it stands to reason that, once the airplane will have overcome most of these difficulties, the airship would be unable to compete unless it can offer on such voyages advantages of another kind. Let us consider the question from this angle. If we take it for granted that both types of aircraft eventually could offer the same safety and reliability in Transatlantic service, two advantages of the airship still remain as a set-off against the greater air speed of the airplane: the greater comfort of the traveler in the airship, especially in view of its ability to fly non-stop, and the *relatively much lower cost of operation.* For the airship must ever remain far behind the airplane as to air speed. Even if the engine power of airships should be augmented and the drag reduced to the furthest possible limits, one could hardly reach a speed of more than 100 m.p.h.

The question which therefore arises is: Will the much greater comfort and convenience offered by the airship appeal to a sufficiently large percentage of the traveling public? Judging by my experience, I do not hesitate to answer this question in the affirmative!

All passengers, without exception, praise the comfort of the accommodations, the smooth and agreeable operation of the airship, the food, and the service.

Advantages of Airship Travel

I am fully aware of the fact that in recent years, especially here in America, great progress has been made in making flying in airplanes more comfortable and agreeable. Noise and vibration have been reduced to a point, which even the most highly strung nerves find possible to endure. And yet, there must always remain a vast difference between travel by airship or flying by airplane. In the airship there is a total absence of either noise or vibration, and passengers have plenty of room to move about at their pleasure, and to sit together and converse normally without being obliged to raise their voices. To these are added the further advantages of excellent food, an attractive smoking room and bar, clear radio reception with musical programs, ship's concerts – we carry a grand piano – and various other entertainments, as for instance games and deck sports, which we expect to develop in time, further to enhance the attraction of airship travel. In this respect, the airship will take a place between the airplane and the modern ocean liner as regards speed on the one hand, and comfort on the other. Nobody has ever seriously considered that the further development of the airplane is ever likely to jeopardize the position of the steamship. There will always be people who, especially during the summer months, will prefer the leisure and low rates of an ocean liner to the speed of aircraft and to whom the speed of the voyage is not the deciding factor. Why then should there not be people who, appreciating the very great amenities and substantial time-saving of a Zeppelin trip, refuse to sacrifice these comforts when some future airplane will be in a position to convey them across the ocean in still less time, but at greater expense?

In this connection the following fact will be of special interest, namely, that in our South American service we already have been able to acquire a number of "steady" guests. These passengers are mostly men and women who cannot spare the time of six to seven weeks necessary for a business trip using steamers plying between South America and Europe. With the airship they can complete the round-trip within three weeks, including a stop-over of two weeks in Brazil or in Europe.

Translated into North Atlantic service that would mean the

following: business men who are pressed for time can leave America and, traveling by airship, reach within 2 days the heart of Europe – let us say, Paris or Frankfurt on Main. They can spend 3 to 4 days, attending to their business, and in another 2½ days be back again, thus accomplishing a business trip in Europe and back again within the short span of one week.

And in my opinion, the question of mail transport will develop along similar lines. I have not the slightest doubt that, in a very few years' time, airplanes will have reached a stage of development, enabling them to carry mail across the North Atlantic perhaps with fuel stops in about 1½ days. Still the carrying capacity of such airplanes will be limited, and the loads they will be able to carry over the ocean must remain restricted. On the other hand, the cost of transoceanic airplane transportation will be considerable, necessitating very high postal rates if a Transatlantic mail service of this kind is expected to pay its own way. The airship, on the other hand, has a capacity to carry 5 or more tons of first-class mail on each voyage, and I was not surprised, therefore, to hear the opinion expressed in high postal circles that in the future there should be three types of Transatlantic postal service:

First, a comparatively slow service, as heretofore, by steamships at a low rate. Second, a very fast mail service by airplane, at a high rate. Third, a fast mail service by airship, at rates which will be very little higher than the present steamship charges.

According to this opinion, one probably could reckon ultimately with a charge of 8 cents for a letter by airship and, in this case, there is no doubt but that a large proportion of all the mail could and should go by airship. This would mean a policy of mail service such as I have always conceived, and the airship would be able to cover a substantial part of its expenses by this means alone.

Greater Economy of Airship

I now come to the second point that I mentioned earlier in the paper, namely, the far greater economy of airship operation as compared with airplane operation. This fact enables the airship to hold its own in competition with the greater speed of the airplane.

At first sight it might appear incredible, but airship operation is relatively inexpensive in comparison to airplane operation. One sees the enormous ships with a comparatively small portion allotted to passenger accommodations. One is surprised at the huge hangars and extensive arrangements at the airport for the accommodation of the ship, and one thinks of the number of men necessary for the landing and mooring of the ship, the general expense of the staff and management, and one comes to the conclusion that airship operation must be extremely costly.

But airships, as we have seen, have the advantage of a very great economical range, and what is not generally known is the fact that the running costs per mile on these long non-stop flights are very low. In theory it is absolutely possible to increase the profits of airship operation to a very surprising degree if one assumes sufficiently large ships carrying capacity payloads on every trip. Such calculations, however, are an idle sport, and only go to prove that airships of increasing size become more and more economical. Therefore, in putting my estimate before you, I shall stick to a basis which takes into account those actual conditions that already exist, and those that can be anticipated reasonably.

I will start with the fact that the airship *Hindenburg* in its ten demonstration trips over the North Atlantic last year, not only earned considerably in excess of out-of-pocket expenses, but amply covered 75 per cent of its total operating costs. In these total costs every conceivable item is included, such as operating expenses, amortization of the ship, insurance, general administration costs, terminal expenses, and so on. These results appear to me very favorable and worthy of attention for the following reasons:

First, because only ten round trips at irregular intervals could be carried out as the *Hindenburg* was obliged at certain given dates to make eight trips to South America.

Second, about 20 per cent of the ship's passenger capacity was devoted to guests of honor, or as you say in the transportation of "dead-heads" – who were invited for reasons of instruction and good will. This represented a loss in revenue of at least 15 per cent, seeing that we had to turn down paying applicants. In view of the large traffic demand we have increased the passenger accommodations on the *Hindenburg* by 40 per cent.

Third, on account of the irregular scheduling of the trips it was impossible to develop a large mail and express volume.

Fourth, the general expenses in America were particularly high, because the organization had to be set up and maintained for only ten round trips.

The total expenses for these ten demonstration voyages approximated $530,000; total revenues were about $400,000, of which $325,000 were passenger revenue and the remainder mail and freight charges. If we had been able to sell the additional 15 per cent of our passenger capacity, to which I have referred, the passenger revenue would have realized an increase of $50,000. Furthermore, we may estimate conservatively, on the basis of a much lower postage rate per ½-oz. letter than the present 40-cent rate, that the airship could count in regular service on at least two tons of first-class mail each one-way trip. This amount represents rather less than 7 per cent of the entire first-class mail which is carried across the ocean to Europe every week. This cargo would mean an increase in mail returns of about $15,000 on each round trip, assuming that the airship received as little as 5 cents on each letter carried, or $150,000 for ten journeys; thus the total expenses would be counterbalanced almost exactly by returns. And do not forget that this estimate is based on those unfavorable expense conditions that governed our ten trial trips, that is, with only one ship in service and this ship charged with all of the general overhead and terminal expenses.

It is necessary, therefore, in striving to obtain an economical North Atlantic airship service, that such service be organized on a sound economic basis, thereby enabling it not only to clear expenses but to become a profitable business.

Instead of 40 paying passengers which we booked per trip last year, we should now be able to count on an average 60 pay-passengers on every trip, in which case passenger revenue would rise from $32,500 to $50,000 per round-trip.

And further, calculating conservatively, one may count on $18,000 worth of mail and freight on each round-trip in the event of a lower postage rate being set, and of the trips being scheduled with such frequency that mail carried by airship would practically always mean a saving of time. This estimate of $18,000 is based on not more than two tons of mail for each trip. It is likely, however, that the postal service will increase enormously and be more than double the value that I have mentioned.

Under these conditions the figures per round-trip would be: passengers, \$50,000; mail and freight, \$18,000, or a round sum of \$68,000, which would mean a profit of \$18,000 per round-trip without counting extras, such as overnight charges levied on passenger luggage, and disregarding possible reductions in operating expenses by an enlarged operating program.

If we can figure on an airship sailing each way every five days, that each airship accomplishes twenty trips yearly, the profit of 20 times \$18,000 would come to \$360,000 per ship. That would work out to be a very handsome return on the capital invested after all other expenses, including each individual ship's share of the terminal charges at either end. And furthermore, I must acknowledge that the development of a future Transatlantic airship service rests entirely on the safety and punctuality with which such service can be carried through. Also much will depend upon the amount of confidence as to its safety with which this new method of transport can inspire the public. A trip in the airship naturally must cease to be an adventure which only "heroes" are prepared to risk. Every normal traveler must feel that he can board the airship with perfect confidence. I would like to call your attention to the fact that I have only counted on an average of 60 passengers on each one-way trip, and only 70 single trips a year. That would mean a total of 4200 passengers a year – a ridiculously low figure when one considers that more than 150,000 passengers paying first-class fares cross the North Atlantic both ways in the course of a year.

In speaking of the necessity for absolute reliability and punctuality, I am not thinking only of the safety of the trip itself, about which I already have expressed myself very fully. I am thinking also especially of the safe and punctual take-off and landing of the airship. In this connection I wish to express my most emphatic and decided opinion that for a punctual and safe airship service, suitable airports are an imperative necessity. They must be situated in meteorologically favorable areas where starting and landing can be accomplished in perfect safety and without great difficulties under almost all weather conditions. This point, I am sorry to say, is often not given sufficient importance. Every city which falls under consideration for traffic reasons, according to my experience, wishes to have an airship harbor in their vicinity, without any thought as to whether its general location is meteorologically suited for this purpose. And apart from this consideration, they want to build their airport in the center of activities, preferably so that the traveler can land conveniently and leave from the roof of his hotel. We have shown in many hundreds of journeys with the *Graf Zeppelin* and the *Hindenburg* that one can land with safety almost anywhere if weather conditions are not too abnormally unfavorable. But it is not only necessary to land the airships, one must be able to house them in a hangar so that the crew and staff can get a rest and do the necessary work on the ship to fit her for her next trip.

And this is the critical point, namely that, if meteorological conditions of an airport are not favorable and if the airport normally is exposed to strong winds from various quarters, great difficulties often are experienced in the handling of the ship when taking it in and out of the hangar. In such cases, one is obliged to remain at the mast and the crew gets no rest and cannot accomplish the necessary work; furthermore, one frequently experiences great delays because it is not possible to get the ship out of the hangar for the scheduled departure. The only solution of all these difficulties, as has been known for many years, is in the rotating hangar. But a rotating hangar costs a lot of money and, in the face of the general skepticism concerning the future of airships, it has been difficult to find the necessary capital to acquire them. Nevertheless, one will be obliged to decide on building rotating hangars because they are the best investment in the long run, in spite of the fact that their cost is twice as much as that of the stationary hangars. A certain equivalent for the rotating hangar, which is the ideal solution, would be a landing station that offers extremely favorable natural meteorological conditions.

Location of Airship Port

The airship port at Lakehurst, N. J., which through the great courtesy of the U. S. Navy we have used up to the present, does not offer the conditions which are needed. There are nearly always – of course, invariably when the airship is there – winds blowing from the wrong quarters so that one has difficulties in getting the ship either in or out of the hangar. I have, therefore, looked about for more suitable landing places and I think I have found such locations in the neighborhood of Baltimore and Washington. It would be a very fortunate solution of this very serious problem if one could decide to build an airship port in the vicinity of these towns. These two cities occupy very central positions, and New York can be reached by airplane or railway respectively in either 1 or 3 hr. I say this advisedly, for I am not of the opinion that New York is the only important traffic consideration, although possibly New Yorkers will not agree with me. Chicago and Detroit are cities with important demands in transportation questions, and they are no farther from Washington or Baltimore than from New York.

I do not need to tell you how happy I would be if the further development of airship transportation, such as I have described, could become a reality, built up with close cooperation between the German Zeppelin Co. and the American Zeppelin Transport Co., which is now going ahead. I am perfectly alive to the fact that on this cooperation between our two countries, the question of "to be or not to be" of a permanent North Atlantic airship service depends.

Here in America there is still a certain amount of opposition to be faced for you have had unfortunate experiences with your "Zeppelins"; you have had very bad luck. Some mistakes also have been made into which I would rather not go now. But mistakes are there to be overcome, and an efficient, energetic people rises above them and refuses to admit that it is licked. As soon as the errors are recognized as such, a great and gifted nation sets to work to rectify them and never says "die." I have always been convinced, and today am more convinced than ever, that America would take up this question again, as soon as it was recognized, as a commercially practicable proposition. And may I venture to think that this claim has been proved by our many years of airship travel and demonstrations.

The proposition is not only a good one, but fortunately there is ample room for further improvement, and I could not wish for a better partner to further this enterprise than the highly developed American technical skill. Obviously Germany has for the moment a certain advantage in this field, having acquired wider experience. But we will be glad to pass our experience on to you, and are convinced that you will add your share of important contributions towards the airship's further development and improvement. It would be particularly gratifying to me if out of those who read this paper stimulating thought and assistance should be forthcoming which will further the cooperation of which I have spoken.

Hypoid Rear-Axle Design and Lubrication

By W. R. Griswold
Chief Research Engineer, Packard Motor Car Co.

CONSIDERABLE effort is made in this paper to explain the nature of hypoid gears and try to convey some idea of why straight mineral oils are not suitable and that special oils are required. The author tells of the effort made by the axle designer and the manufacturer to attain, as nearly as possible, perfection in building the axles, pointing out that all the benefit of this work can be lost by the use of an unsuitable lubricant.

Hypoid gears and their tooth action are compared with preceding types. Gear mountings are described, explaining the advantages of making them rigid. The influence of the lubricant used on load capacity and wear of hypoid gears is explained in considerable detail. Characteristics of suitable E.P. lubricants are discussed with recommendations for their proper selection.

THE recent widespread adoption of hypoid rear-axle gears brings about, quite naturally, a widespread interest in the nature of hypoid gears, their behavior, the peculiarities associated in the design of axles in which they are incorporated, their behavior in service, and the nature of the lubrication requirements.[1] By peculiar circumstances, the author has been associated with the design and development of hypoid gears since their very inception, in fact, his experience began with the design of the very first hypoid-gear axle of which there is any record.

Some twelve or thirteen years ago, after many attempts to improve on the spiral-bevel gear, the Gleason Works succeeded in making a pair of hypoid gears. F. E. McMullen brought that first pair of gears to Col. J. G. Vincent, vice-president of Packard who, immediately recognizing the inherent merits of the hypoid principle, proceeded to design a special axle to accommodate this form of gearing. At that time the Model T Ford used straight-tooth bevel gears, and spiral-bevel gears had been in general use by other manufacturers for a comparatively few years. A year or two later Packard adopted hypoid gears in regular production and has not made any spiral-bevel gear axles since.

Research, development, and testing work conducted through the years since, much of it in collaboration with the Gleason Works, and more than ten years' field experience with Packard cars, have yielded much which has become, to a large extent, a part of the technical knowledge in our great industry. Like many of the other complex mechanisms with which we find ourselves in daily contact, much of this knowledge is incorporated in existing successful designs which are the products of an evolution during which changes and improvements have come slowly and sometimes painfully, rather than as classified technical knowledge in the minds of engineers. For this reason those who only recently have become associated with the design, manufacturing, or service phases of hypoid axles, will have much to learn in properly evaluating the importance of specific items and their relations to one another and, naturally, there exists much divergence of opinion and many contradictory notions.

This paper, therefore, is presented to cover as comprehensively as possible all of the important experience obtained in the last ten or twelve years but as concisely as will be permitted by the subject matter. Naturally, emphasis will be placed on some elements which may appear to be repetition

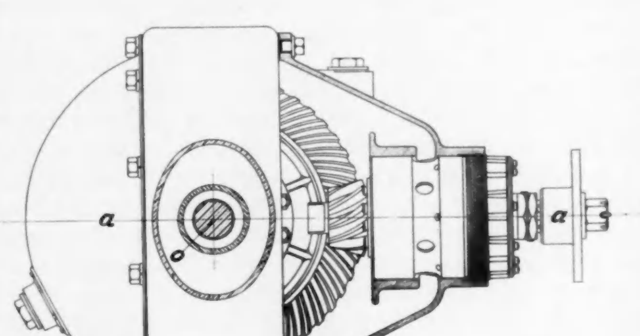


Fig. 1 – Spiral-Bevel Rear Axle with Overhung Pinion Mounting

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1937.]

[1] See Proceedings, Sixth Mid-Year Meeting, American Petroleum Institute, May, 1936, pp. 7-19; "Rear-Axle Design and Lubrication," by W. R. Griswold.

or about which some of you may presume to have complete knowledge.

Up to within this past year the use of hypoid rear axles had been confined to but a few manufacturers, and so many of the influences, commercial or otherwise, which had to do with their use, were absent or were handled easily by the individual manufacturer. Packard, who has for more than ten years been using hypoid axles exclusively in regular production, has enjoyed the position during that entire period (until the introduction of the Packard "120") of building cars in only the higher priced brackets which were distributed to a clientele who demanded the best and most expert service and who were inclined to regard service instruction with the fervor of those who appreciate the value of honest advice for their own best interests. For this reason Packard cars, very nearly to the extent of 100 per cent, were serviced for rear-axle lubrication in Packard service stations. This picture has been largely true on the Packard "120" until recent months when hypoid axles by volume producers have been introduced, particularly in the lower price brackets.

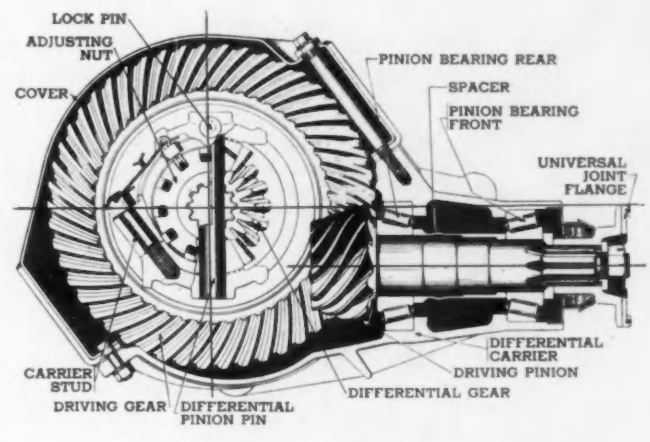


Fig. 3 – Hypoid Rear Axle with Overhung Pinion Mounting

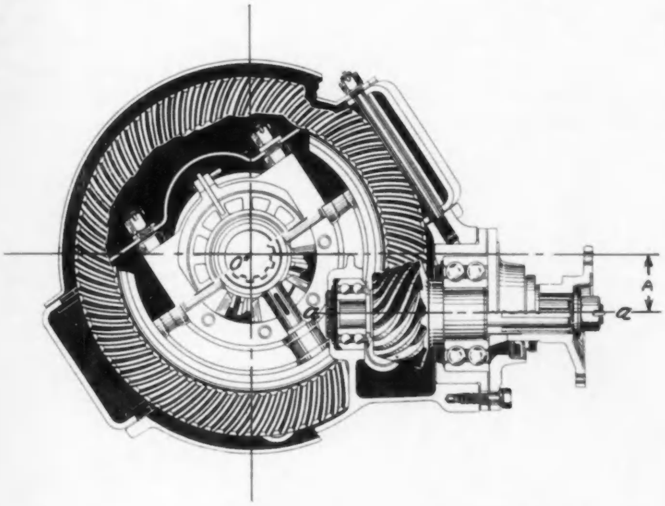


Fig. 2 – Hypoid Rear Axle with Straddle-Mounted Pinion

It is well known that hypoid gears require special lubrication and, of course, the widespread adoption of hypoid gears creates a demand for the special type of lubricant required and for a widespread distribution of this lubricant. The promise of large volume business in hypoid lubricants naturally has awakened a definite interest among lubricant manufacturers who wish to participate in this business. This activity has brought about the introduction of new types of hypoid lubricants and, consequently, has focused much attention on the problems of the distribution of these lubricants and the servicing of hypoid axles in the field. Unfortunately, not all who are engaged in the production and servicing of lubricants are equipped with the facilities or the technical personnel required to produce a satisfactory product and to maintain the necessary uniformity and, much as I regret to say it, some are so zealous to do business and are so indifferent to the best interests of their customers as to push the sale of lubricants that are entirely unsatisfactory, or even ruinously injurious when used in hypoid axles. Although to a small extent there may be some cause to suspect that inferior lubricants are being sold with deliberate intent, there are more cases where they are sold through ignorance of the requirements of these lubricants and because of a woefully inadequate knowledge of the merits and behavior of the lubricant by its producer. In some few cases the producer has failed by believing that his product possessed exceptionally high merit, and accounted for trouble by blaming the units in which it was used. Therefore, much of what is to be presented here is intended to clear up some of these misconceptions and to point out the great need for caution on the part of the makers and users of hypoid lubricants, in order to obtain long and satisfactory life from hypoid axles.

Comparison of Gear Types

Although hypoid gears have been used in regular production for more than ten years, I am asked frequently what a hypoid gear is and how it differs from the older spiral-bevel gear. This best can be explained by referring to Figs. 1 and 2 which show respectively a spiral-bevel construction and a hypoid design. In Fig. 1 the axis of the ring gear is at O and the axis of the pinion (a-a) is disposed perpendicularly to the axis of the gear through O. In Fig. 2 the gear axis is perpendicular to the pinion axis (a-a), but it is observed that the axes do not intersect but are displaced by the distance A which is called the hypoid offset. Pinion mountings may be of two types: the straddle type shown in Fig. 2 or the overhung type shown in Fig. 3.

Both bevel and hypoid gearing are based fundamentally on pitch surfaces which rotate about their axes with a uniform velocity ratio. In the case of the spiral-bevel gears these pitch surfaces are cones, as shown in Fig. 4, which obviously may rotate about their axes with pure rolling motion on the contacting surface. For gears operating about oblique but non-intersecting axes at a constant velocity ratio, the pitch surfaces also must be surfaces of revolution, but the enveloping elements form a hyperbolic curve. Such pitch surfaces are known as hyperboloids of revolution which, for brevity, are called hypoids, shown in Fig. 5. Only a portion of the hypoid surface is required in the formation of the gears so that, in appearance, the pitch surfaces approximate sections of cones.

For toothed gears an obvious requirement is that the pitch spacing of the teeth must be constant in order that teeth register properly with mating teeth at each revolution. This requirement is met in bevel gears with straight-line cone elements along which the cones are tangent. The hypoid ele-

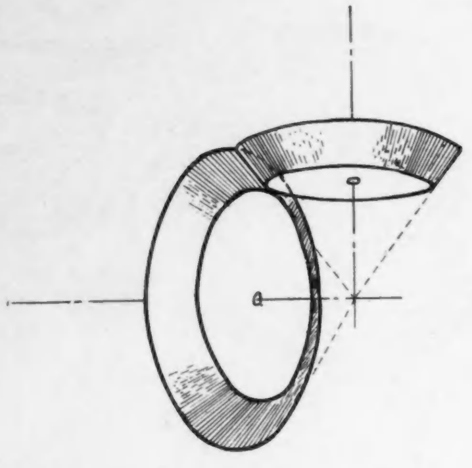


Fig. 4 – Pitch Surfaces for Bevel Gears Are Cones

ments also are straight and the hypoid pitch surfaces are tangent along one rectilinear element.

The essential difference between the cone surfaces and the hypoid surfaces is that, in the former, the action between the two cones is pure rolling with tangency along a rectilinear conical element whereas, in the latter, the pitch surfaces roll and slide on one another as they rotate with a constant velocity ratio. The tangency is along a rectilinear element, and the motion between the two is composed of pure rolling perpendicular to the element, and pure sliding longitudinally along the element. It is also evident that as A, the pinion offset, is diminished the pitch surfaces approach nearer and nearer to conical surfaces.

If now we put teeth on these two types of pitch surfaces providing for the proper tooth-to-tooth spacing and tooth profiles of such shape as to meet the conditions of constant velocity ratio, we get respectively bevel gears or skew bevel gears. If we make the teeth straight, following the pitch element of the surfaces, we will have respectively straight-tooth bevel gears or skew bevel gears. If we make the tooth elements curved, such as circular arcs, and displace them so that tangents to their mid points make an angle to the pitch surface elements we get what we familiarly know as spiral-bevel gears or hypoid gears respectively.

[2] Two types of profiles provide for constant velocity ratio, the cycloidal and the involute. The cycloidal type has become obsolete.
[3] "Involute Spur Gears," by Earle Buckingham, p. 40, Fig. 21.

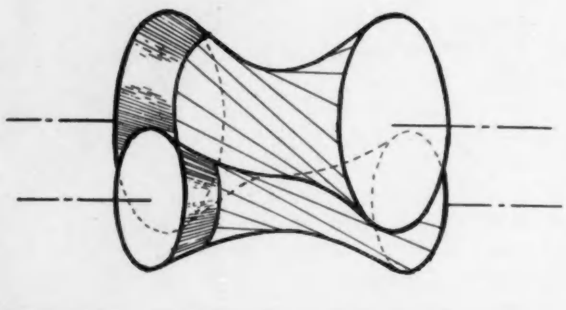


Fig. 5 – Pitch Surfaces for Hypoid Gears Are Hyperboloids of Revolution

Comparison of Tooth Action

A large part of the fundamental knowledge of the gear engineer is the kinematics of tooth action during the meshing cycle.

The most elemental action between two meshing teeth occurs in ordinary spur gears, the pitch surfaces of which are cylinders with parallel axes and the velocity ratio of which is equal to the ratio of their diameters. The action between two meshing involute[2] teeth is not pure rolling but a mixture of rolling and sliding, the ratio of slide to roll changing throughout the meshing interval. This action is demonstrated in Figs. 6 and 7. In Fig. 6 the right half shows two contacting tooth profiles with portions shaded to show the relation of sliding action to rolling action. For instance, the portion (7) of the right-hand gear contacts with the relatively narrow portion (1) of the left-hand gear. Since these two portions go through contact in simultaneously equal time periods, it is evident that they must slide on one another, and the amount

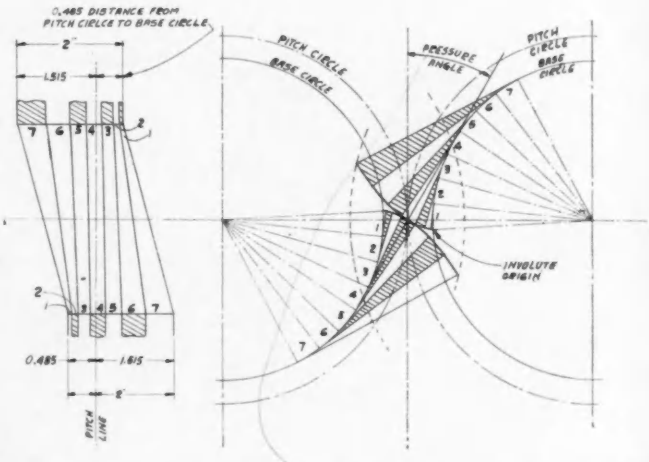


Fig. 6 – Sliding Action of Involute Profiles of a Pair of Meshing Gear Teeth

of sliding is a function of the difference in the profile lengths. For this portion of contact the sliding is high as compared with the portions (4-4) which are of equal length and, consequently, indicates pure rolling action in the neighborhood of the pitch line. The left-hand portion of Fig. 6 shows the relative profile engagements laid out on a flat plane, the contacts of the portions being as follows: (7) with (1), (6) with (2), (5) with (3), (4) with (4), (3) with (5), (2) with (6), and (1) with (7). It is seen easily that the sliding is greatest at the root of the tooth, becomes zero at the pitch line, and high again at the end of the tooth.

The relative sliding at different points on the profile is shown better by the sliding diagrams in Fig. 7,[3] the factor expressing the ratio of the slide to roll. All the diagrams pass through zero at the pitch line which, of course, is to be expected since the sliding is zero at that point. These diagrams show how it is possible to control the extremes of slide-roll ratio by varying the properties of the tooth length above and below the pitch line. You observe that the ratio becomes infinity for the old Brown and Sharp proportions and is quite moderate in the Maag proportions.

The involute tooth action of spiral-bevel gears is only slightly different from that in spur gears (cylindrical gears). A common approximation is to consider the tooth action to be that of equivalent spur gears, the diameters of which are equal

to twice the length of the normals from the pitch cone element to the gear axis. Actually, however, the tooth action at points of contact other than at the pitch line involves a slight longitudinal sliding action along the tooth. This sliding ordinarily is so small as to be regarded as of negligible importance, and the spiral shape or curvature of the tooth has little influence.

The tooth action in hypoid gears involves, in addition to the involute sliding action, a relative sliding motion longitudinally along the teeth as indicated in Fig. 8. Here the slide due to involute action is represented by the vector *a* and the longitudinal slide due to the hypoid offset is represented by *b*, the resultant sliding being shown by *c*. The angle changes throughout the meshing cycle and varies in magnitude, being minimum at the pitch line and maximum at the root and addendum of the tooth. A comparison of relative absolute sliding values for spiral-bevel and for hypoid tooth action is shown in Fig. 9[4] and reveals two essential facts: (1) that the absolute value of the maximums for the hypoid is not much greater than for the spiral bevel, and (2) that the minimum values occur at the pitch line, the theoretical value being zero for the spiral-bevel gears.

The tooth action, that is as to sliding, can be varied to some extent by the choice of pitch, tooth heights, pressure angles, long and short addendum proportions for both types of gearing but, for the hypoid, the pinion offset is a most important factor in affecting slide.

Tooth Pressures

The instantaneous surface pressures occurring in the contact zones of a pair of meshing teeth are high and are affected by many variables. Gear men, of course, understand that the no-load contacts are lines of infinitesimal width and

[4] "The Lubrication Requirements of Automotive Rear Axles," by H. R. Wolf, Proceedings, Seventeenth Annual Meeting, American Petroleum Institute, Section III, p. 33, Fig. 3.

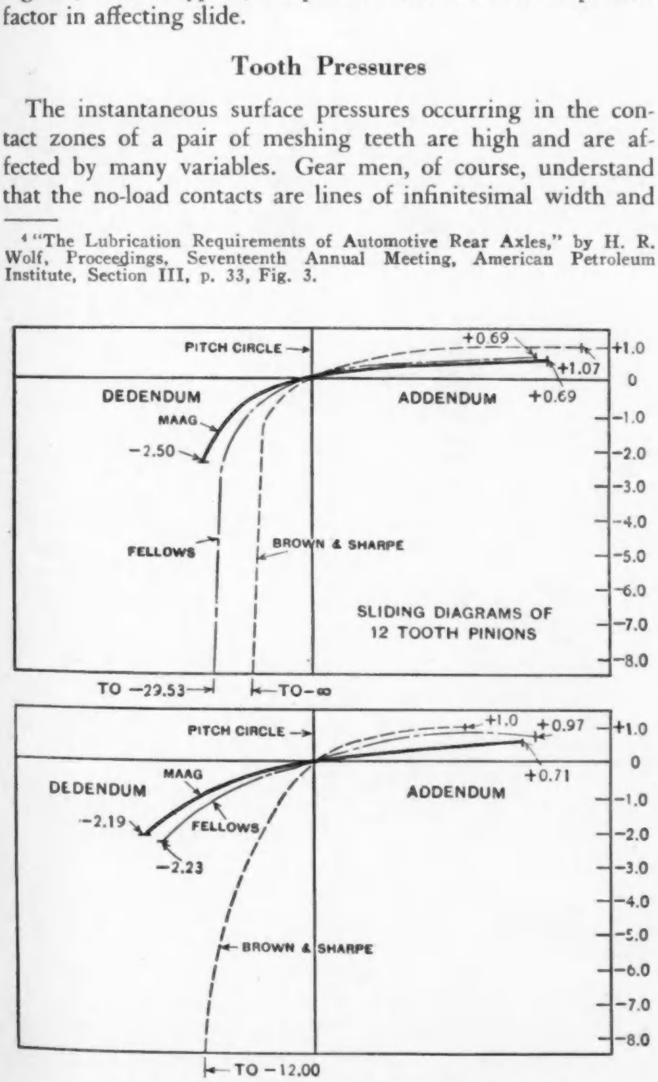


Fig. 7 – Sliding Diagram of Gear-Tooth Profile

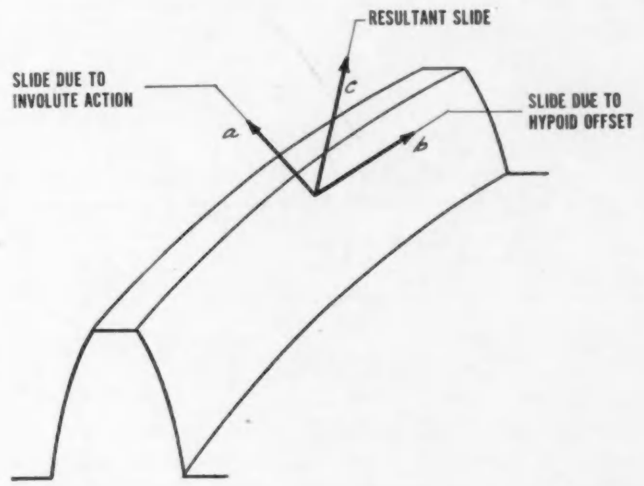


Fig. 8 – Resultant Sliding Action on Hypoid Pinion Tooth Due To Combination of Involute Action and Hypoid Action

occur simultaneously on several teeth assuming, of course, perfect tooth spacing. Under load, mutual compression takes place at the contacts and the lines become areas of finite width, of a dimension that is very narrow as compared to the working surface of the teeth. The distribution of load among all the pairs of teeth in contact is quite indeterminate and depends upon the elastic deformation, both general and local, of the gear teeth, the relative curvature of the tooth surfaces in contact, the thickness of the tooth, the number of pairs of teeth in contact, the deformation of the gear mountings permitting relative displacement of the gear and pinion, the pressure and spiral angles and, of course, the load. Even if we could assume that the mountings be perfectly rigid, the load distribution among the teeth will not be anywhere near equal, as the pair in mesh nearest the pitch line will be the most heavily loaded.

The maximum tooth loading of the hypoid-gear set will be less than the tooth loading in a spiral-bevel-gear set of the same ring gear diameter and with the same number of teeth in gears and pinions because there are more pairs of teeth in contact. The tooth loading, however, is not reduced in direct proportion to the increase in number of pairs of teeth in contact because the contacts vary in position along the tooth and as to location above and below the pitch line. However, the gear designer has some latitude in choosing the proportions of the gear teeth in order to obtain the minimum maximum pressure. Also, because of the inherently stronger teeth of the hypoid pinion, finer pitches may be used yet staying within the requirements of tooth strength against rupture or tooth breakage. But, with too fine pitches, the tooth deformation may become so great as to offset the advantage of having a larger number of pairs of teeth in contact. The tooth load distribution, however, is altered considerably by the deformation of the carrier or mounting for the bearings, deformation of the bearings themselves, and deformation in the pinion shaft and in the differential case on which the gear is mounted.

Gear Mounting

The rigidity of the gear mountings not only affects the pressures occurring at the contacting zones but also is a determining factor in the quietness of operation of the gears.

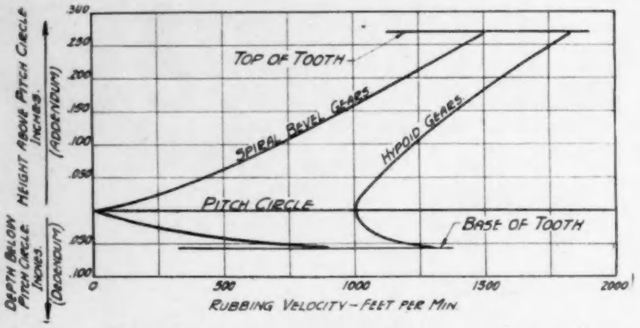


Fig. 9 – Relative Tooth-Surface Rubbing Velocities for Spiral-Bevel and Hypoid Rear-Axle Gears

Referring to Fig. 10, the full lines indicate the normal relationship of gear and pinion for a straddle-mounted pinion design. The dotted outline indicates, much exaggerated, the displaced position of the pinion when subjected to forward driving torque. The pinion is displaced upward through the angle θ and actually away from the gear by the distance X.

In Fig. 11 is shown the relative displacement in an overhung mounting and, in Fig. 12, full lines again show the normal position of gear and pinion and the dotted lines show that the displacement of the gear and pinion is away from each other. In reverse drive, similar deflections occur, excepting that the vertical displacements are reversed as are also the axial thrust displacements of the pinion. Displacements, that is the important ones, usually are classified as follows: (1) pinion lift, (2) pinion side movement, (3) pinion axial movement, (4) gear lift, and (5) gear side movement.

These displacements actually are determined by operating the axle unit in a test machine adapted to apply all load variations from no load to the maximum load to which the unit is expected to operate at the full motor torque in low and reverse gears in the car. Micrometer dial indicators are mounted at vital points and the displacements determined by loading the unit with the gears running very slowly; at the same time the gear teeth are painted with red lead at the various loads and a pattern of the tooth contact is determined for these loads. In this way it is possible to correlate very accurately the change in tooth-contact pattern with the displacements which occur at the various loadings.

With the indicators mounted at vital points it is also possible to determine what factors contribute to the deflection. The deflections in the bearings alone can be determined, and the individual deflections occurring in the ring gear, differential bearing pedestals, and in the differential case, can all be measured. This condition is also true with the pinion mounting in which the deflections occurring in the pinion bearings can be separated from the deflections occurring in the pinion shaft and in the housing. It is by this means that gear sections, pinion shaft diameter, bearing sizes, and the structural proportion of the differential case and carrier can be arrived at for the maximum deflection values into which a good design would fall. In Fig. 11 is shown the relative displacement for an overhung pinion, and in Fig. 12 is shown the relative displacement of the gear and pinion in the plan view.

Obviously, since all of these materials are elastic in character, it is impossible to make the mountings perfectly rigid. The maximum displacements that are permissible are determined or dictated largely by limitations in contacting patterns of the gear teeth.

In Fig. 13 are shown contact patterns of the well-designed axle unit. The variation in tooth pattern from no load to full load is caused by the relative displacement on gear and pinion due to deflections and mounting and, to some extent, to the deformation of the gear teeth themselves. One criterion for good design is that, under full-load conditions, the contact pattern must not show high pressures at the heel of the tooth or shall not show a heavy wipe at the top or bottom of the tooth. With the proper tooth pattern under maximum load conditions, the displacements must not be great enough to yield a too-short pattern under no-load conditions. If the tooth pattern is too short, the gears will be noisy under so-called floating loads or at very light loads in drive or coast. If the displacements are too great and the gear and pinion are lapped for sufficiently long tooth-contact pattern under light-load conditions in order to avoid noise then, when maximum loads are applied, the tooth patterns will show concentrated loadings which, of course, means that the capacity of the unit is reduced because of the high unit pressures existing. In addi-

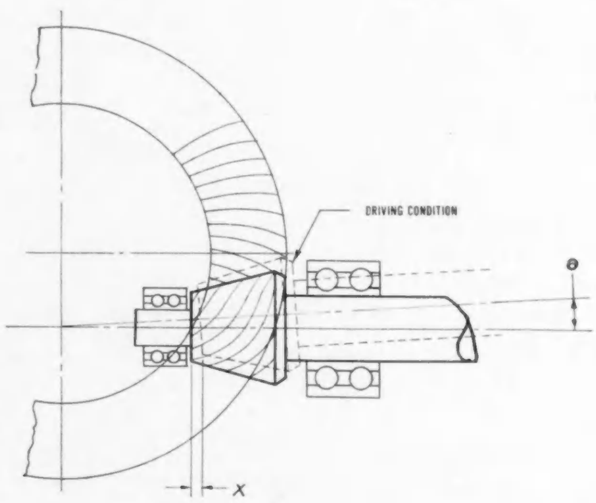


Fig. 10 – Exaggerated Displacement of a Straddle-Mounted Pinion Under Load

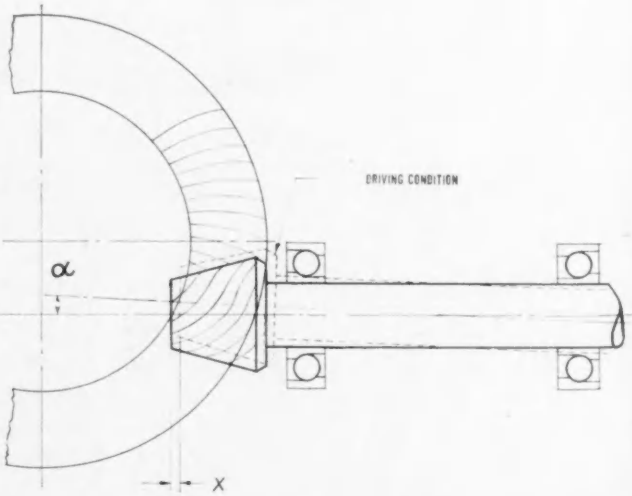


Fig. 11 – Exaggerated Displacement of an Overhung Pinion Under Load

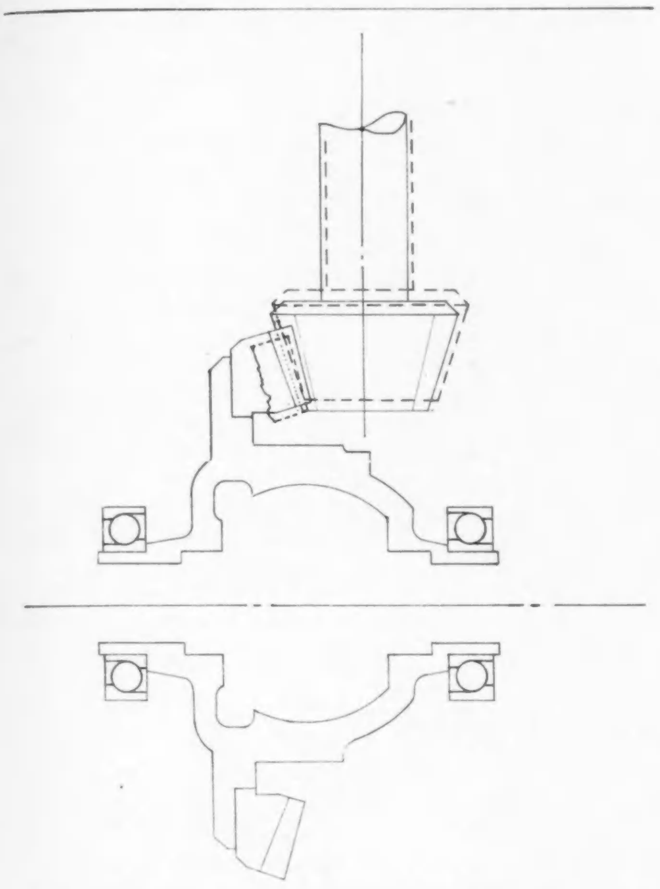


Fig. 12 – Relative Displacement of Gear and Pinion in Plan View

tion to gear and pinion displacements caused by the deformation of parts, it is obviously a matter of great importance to eliminate lost motion in the bearings and, for this reason, preloading of bearings is essential.

Gears and pinions are lapped to a high precision during their manufacture and are lapped to the backlash to which the gear and pinion are intended to run in the axle. The gear and pinion, for instance, which have a proper tooth contact at a backlash of 0.003 in., will show a marked different pattern at 0.006-in. backlash and, if the gear and pinion were adjusted to give a proper bearing on one side of the tooth with the 0.006-in. backlash, then the contact on the opposite side of the teeth would be out of position sufficiently to cause noise at light loadings and concentrated tooth pressures under maximum load.

Preloading of bearings has the advantage of reducing the deformation displacements after the looseness is taken up, since the yield varies in rate, being greatest at low loads. It is obvious that initial seating-in or wear in the bearings must be reduced to such low magnitudes that the displacement of the gear and pinion will not be affected. The general practice for many years has been to preload the differential bearings by screwing up the adjusting nuts so as to actually cause a slight spread at the bearing caps. This elastic yield is utilized in maintaining an appreciable axial loading on the bearings. For pinion mountings, bearing preloading has not been general, but it is now universal practice to have some preload, and in some types of design it is common to have from a 1000- to 1500-lb. axial preload.

Much effort has been expended in order to limit the yield and looseness in bearings. Great improvement has been made to eliminate the seating-in of new bearings which has been accomplished by high precision in manufacture and by surface finishes of extraordinary smoothness. In ball bearings it is common practice to have highly lapped finishes on the balls which are made to extraordinary precision for size and roundness, and to form the grooves accurately and polish them to a mirror-like finish. In taper-roller bearings for pinion mountings, lapped rolls and mirror-finish race surfaces are demanded. Years of effort have been expended on the design and manufacture of bearings that will deform the least possible amount under load. This purpose has been accomplished to a large extent by refinements in design and specifications for great precision.

As I have mentioned, after heat-treatment the gears are lapped to a smooth finish and to a high precision of tooth-to-tooth spacing. Before the gears are assembled in the axle units, they are tested on machines in accurate running relationship. In a well-built axle unit this relationship is maintained in the assembly, which is possible only when the parts on which the gears are mounted are made accurately and the carrier is machined accurately. The limits of accuracy on parallelism, squareness, and center distances of the gear axes in the machining of the parts affecting the mounting of the gears, are extraordinarily close, and the cone-distance positions of the gear and pinion are maintained to produce in the carrier the tooth patterns that are exhibited on the test machine after lapping. For most of these positions the permissible limit is 0.001 in., and the maximum limit on cone position for the pinion in a well-made unit will not exceed 0.0015 in.

Precision in manufacturing is now rapidly approaching the limitations of the measuring fixtures that can be regarded as

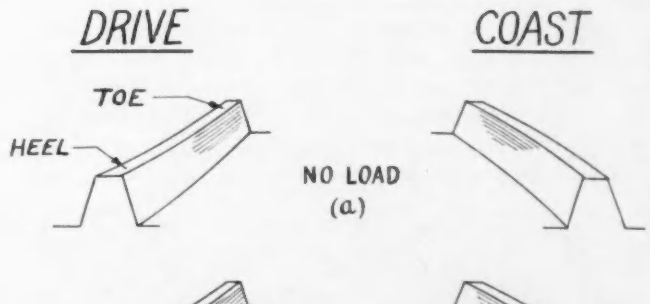


Fig. 13 – Contacts on Gear Teeth for Various Loads

falling within the category of practical production equipment. To obtain this high precision in manufacture has not been easy, but the realization of the need for it has been demonstrated well by experience, and its accomplishment has been possible only by the fullest cooperation of manufacturing and production talent, and by the readiness of managements to invest heavily in the most modern equipment procurable.

Lubrication

We come now to the subject of lubrication and, in any discussion of this subject, we are forced to accept the concept that the lubricant is just as much a structural element in the gear unit as the steel in the gears and bearings, and that lubrication is just as complete a functional element as is that of the teeth on the gears; the unit will just not operate with the absence of either element.

The interdependence of the structural elements is manifest in the capacity of the unit to transmit power. If the load capacity of the lubricant is low, the capacity of the unit is limited to its break-down value; if the strength of the metal parts is low, regardless of the load capacity of the lubricant, then the power capacity of the gear unit is limited to the rupture strength of the metal. I realize that this concept is not new, but sometimes the reiteration of a fundamental truth brings our objectives into a clearer focus.

It hardly should be necessary to say that in recent years the capacity of axle units has increased beyond the load capacity of straight mineral oils which for so many years have been regarded as the maximum standard for lubrication value. In other words, axle capacity has been limited by the lubricating ability of straight mineral oils to lubricate primarily the gear teeth. Such structural elements as the housings, bearing mountings, the bearings themselves, and those parts on which the gears are mounted, have been increased in capacity to transmit power beyond the load-carrying capacities of the lubricant to lubricate the gear teeth properly and, not only has the ability to reduce axle size or to increase power-transmitting capacity been measured by the limits of lubricants, but this situation has prevented the thought that any such thing as increasing the load capacity of lubricants could possibly exist. But the achievement of getting more and more power from engines without increasing their size and the continuous effort to lower roof lines and floor height of automobiles has brought about the necessity of making smaller axles transmit larger amounts of power successfully.

With this necessity has evaporated the compunctions about using special lubricants, and the question is now not whether a special lubricant should be used but what kind of special lubricants can be used and how the quality and distribution can be controlled. It seems to be out of the question to ask the car owner to see to it that he does get the proper lubricant for his axle. He is told in the instruction book to obtain lubricant only from an authorized service station for his car, on the other hand, he is told over the radio to buy this and that make of lubricant, and he is told by the man from whom he buys his gasoline that he has a special brand of rear-axle lubricant that is much better than can be obtained through authorized service. Whom is he to believe?

It seems unfair and un-American to restrict the development of lubricants by a policy reciting arbitrary specifications either on the part of the automobile manufacturer or by special groups. The answer to this question is not readily apparent so long as there exists confusion involved by distribution problems and ignorance concerning the true requirements of lubrication.

Proper lubrication concerns both the gears and bearings and should be related to the requirements imposed by normal service of the axle unit and not by abnormally severe demands imposed by arbitrary tests. The impracticability of having one type of lubricant for the gears and another for the bearings is quite obvious but, since the lubrication for the bearings is dictated by the requirements for the gears, this lubricant shall not be harmful to the bearings. What constitutes proper lubrication and what is a proper lubricant may be stated simply:

For ordinary normal service we must have:

(1) A lubricant that will provide a film between gear teeth to prevent the gears from scoring at all load, speed, and temperature conditions.

(2) That the lubricant will not cause of itself or contribute to wear of the gear and bearing parts.

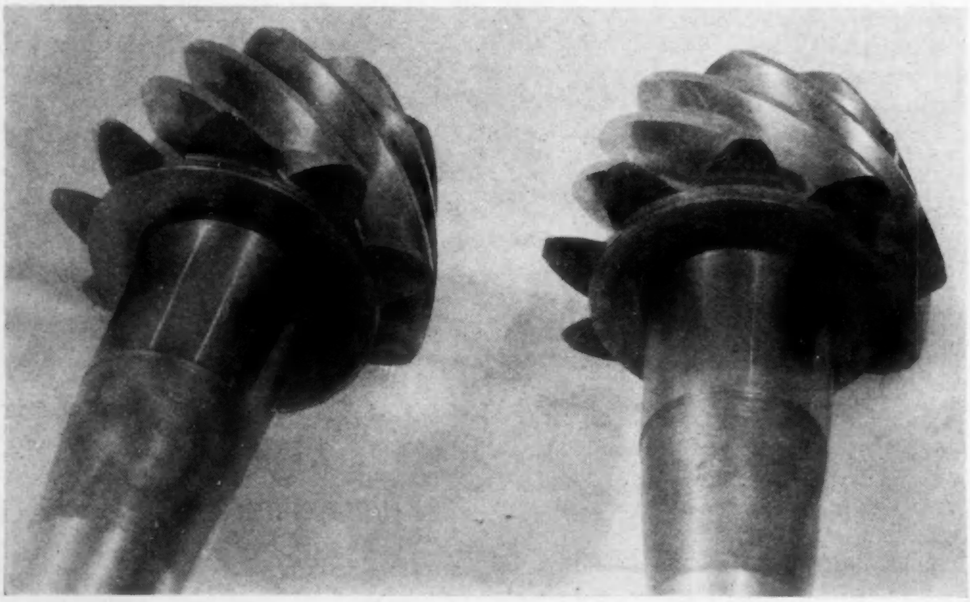


Fig. 14 – Polished Surfaces After Test with Satisfactory Lubricant

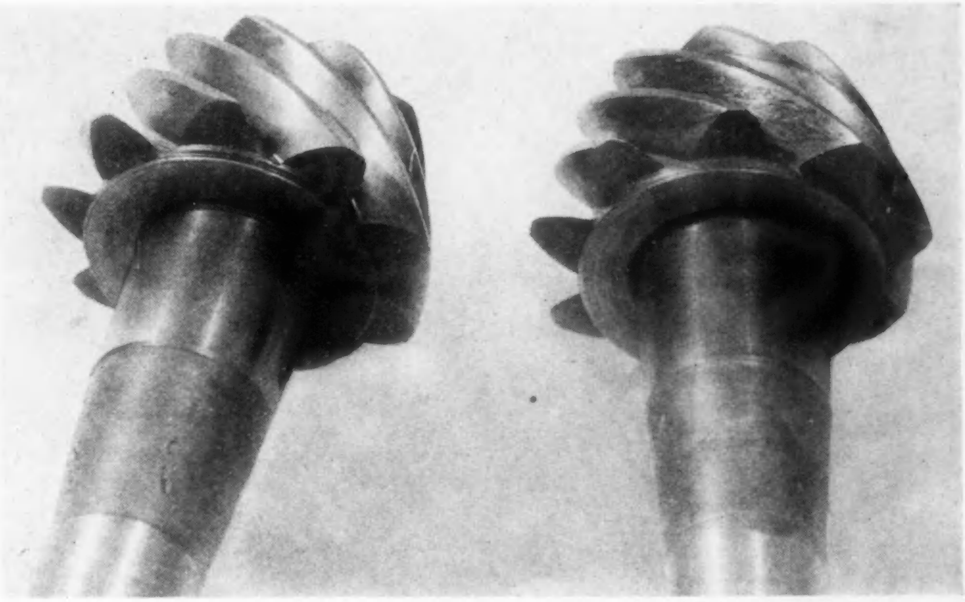


Fig. 15 – Light Score After Testing with Low E.P. Lubricant

Fig. 16 – Heavy Score After Test with Lead-Soap Lubricant That Had Deteriorated After Standing

(3) That the lubricant shall have no harmful chemical activity.

Now it is apparent in analyzing the three requirements that a lubricant to fulfill the prerequisites of item (1) shall not suffer any diminution of load capacity, shall not suffer any destructive effects from temperature or agitation, and certainly shall not be affected by whatever influences that atmospheric conditions may have. In other words, we find here that the lubricant must be stable during use and under the conditions with which it is used at least so far as it affects load-carrying capacities. To meet the prerequisites of item (2) over a reasonable period of time, the lubricant must be non-abrasive at the beginning and it shall not develop any abrasiveness during its use. To meet the requirements of item (3), the lubricant must have no chemical activity which will deteriorate the material normally used in axle construction. A more exacting desirability is that it possess no chemical effect beyond that existing with straight mineral oils.

We might reasonably analyze more fully these three prerequisites, as follows:

Load Capacity

The load capacity of the lubricant is necessarily a prerequisite established by the size of the axle and its power-transmitting capacity and the service to which it is subjected, the last of which only needs elaboration. What is meant by ordinary normal service is that imposed by operating the car at what might be reasonably experienced in the course of use even including those periods of extraordinary severe usage demanded in exigencies. From the designer's point of view the gear capacity is rated easily and with sufficient accuracy by the tooth loading per inch of face.

So far as load capacity of the lubricant is concerned, I am told that this property may be measured by the so-called "E.P." value, and I have been warned that this may mean either high E.P. index with true oily-film lubrication or that it may mean a measure of the anti-welding properties of a solid film.

I do not propose to become involved in a controversy that may exist between lubrication experts. From the standpoint of the axle, the only thing of importance is that the load-carrying capacity shall be such as to prevent scoring and that the lubricant shall have no other detrimental effects. That there must be no other detrimental effects is most important. In considering the load capacity, all reference must not be directed to fresh lubricant; the load capacity after 10,000 or more miles is just as important as its capacity before use. Its capacity to carry load in summer or winter, and after long storage either in the factory, the service station, or in the axle, is of paramount importance.

For the measure of load capacity we have adequate testing facilities for the initial approval of a lubricant and, so long as arbitrary conditions are established for the testing, a fair index of the load capacity is obtainable; but, to determine the stability of a lubricant in use over a period of time, is a long process and an expensive one and too frequently impossible of duplication in all variables. Whatever may be the high-load capacity of a lubricant, unless it holds up during use, the axle designer cannot make full use of it, and any great reduction in load capacity is certain to be an indication of other forms of deterioration of the lubricant. There are various methods for rating the load capacity of lubricants, but not all laboratory tests are of value unless they accurately correlate tests of production axle units and road experience.

There are various methods of determining load capacity of the lubricant by testing in full-sized axles. A common test is to engage the clutch suddenly with the car running at various speeds in either direct drive or second speed while the motor is idling. This is a type of test that can be made as severe as desired. Caution should be exercised in making this type of test so that the severity shall not be greatly beyond what might be expected in normal service. In Figs. 14 to 18 are shown various results on the pinion teeth using various types of lubricant.

A much more reliable test and one which we have used for many years is to run the axle for 9 hr. on the drive side at full motor torque in second gear and for 1 hr. in reverse at the same load, maintaining some sort of artificial cooling, generally that obtained from circulating fans. Such a test gives more reliable information as to the items of stability, abrasiveness, and chemical activity as well as to the load-carrying capacity of the lubricant.

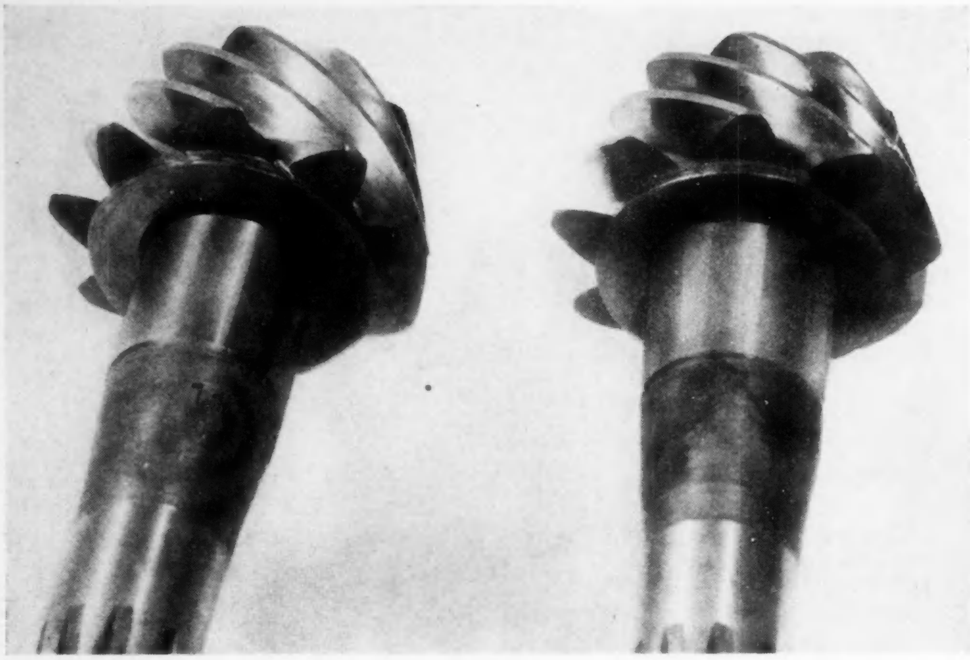


Fig. 17 – Light Score Due To Low E.P. Values

Fig. 18 – Another Example of Good Lubrication On Score Test

Wear

I have on other occasions used the term abrasiveness to describe the influence of the lubricant on wear of axle parts and have been warned that this is not a proper term as E.P. lubricants, so-called, are not abrasive. Here again is a controversial point of interest only to lubricant technologists and, whether the lubricant is of itself abrasive or whether through some faulty mechanism in its operation it causes wear in the axle parts, the terminology is unimportant. The net result is undesirable and harmful. The postulation that E.P. lubricants are not abrasive is not susceptible to proof. On the contrary, there is much evidence to indicate that many of them are abrasive, whether they start out in the beginning that way or whether they become abrasive during use. The facts that axles wear more with some lubricants than with others, and more in summer than in winter are to me incontrovertible proof that some lubricants are abrasive, and some more abrasive than others.

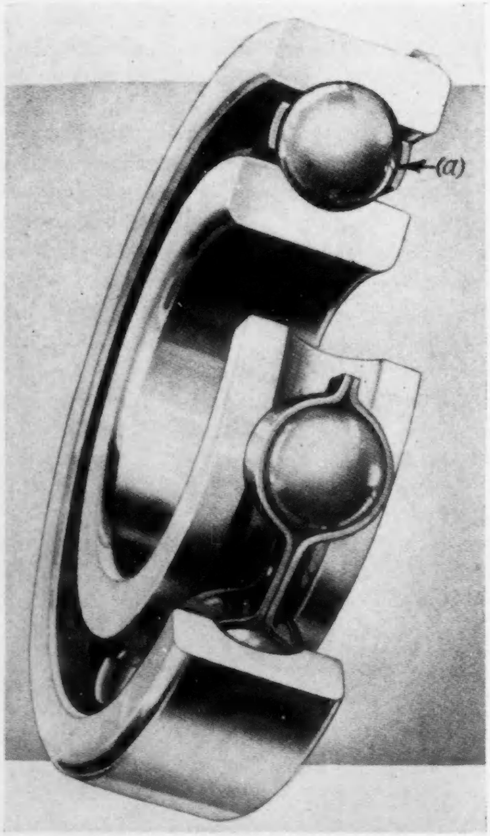


Fig. 19 – High Pressure Between Ball and Cage Causes Lapping Action On Balls When Run in Abrasive Lubricants

To begin with, there is no such thing as pure rolling motion in any of the elements making up an axle assembly, and wear will occur whenever sliding occurs under pressure in the presence of an abrasive. We are not only interested in the wear of the gear teeth, but also primarily so in the wear in bearings, and it is to explain how important the wear in bearings is that I took so much trouble previously to explain the great care taken in the high precision to which bearings are fitted and the gears themselves are mounted. Any wear in the gear teeth or in the bearings starts a vicious circle of wear.

It is probably self-evident why the gear teeth should wear in the presence of an abrasive. But the wear in ball and roller bearings might be somewhat obscured by the fact that these parts are presumed to operate with pure rolling motion. It is impossible to prevent sliding no matter how geometrically accurate the parts may be made originally because the parts deform under load and produce zones of sliding under extremely high pressures. There are other zones of considerable sliding under pressure that require non-abrasive lubrication to avoid wear.

Ball bearings are subject to wear because of pressure and sliding existing between the ball and cage or separator. In Fig. 19 the point (*a*) is a zone of pressure between ball and cage caused by combined radial and thrust loads on the bearing. Since the relative motion between the two is sliding, the presence of an abrasive causes wear and is exhibited primarily in lapping the ball down to a smaller size. I want to mention here that the balls in the bearings, as used in motor cars and

demanded by specifications, which do not have an accuracy for diameter and sphericity within 25 millionths of an inch are rejected. In taper-roller bearings wear caused by abrasive occurs at two points. In Fig. 20 it will be seen that at (*a*) the end of the roll contacts a thrust rib on the inner race. This thrust rib is necessary to guide the roller and is subjected to a considerable thrust from the roller. As all of the relative motion between the end of the roller and the rib is sliding, the contacting surfaces are subject to wear in the presence of an abrasive. At (*b*) the surface of the roller contacts the case which maintains the spacing between the rollers, and all relative motion between roller and case is sliding. Fig. 21 shows a failure of a taper-roller bearing due to lack of lubricant.

If wear occurs at either point the "stand" of the bearing is decreased, which means that the inner race approaches axially further into the outer race thereby developing end-wise clearance. If two sets of bearings are used to oppose thrust in opposite directions, the clearance developed produces an endwise looseness or chuck when the thrusts reverse.

The influences of the lubricant on the wear in the axle are measured best by long-time tests with a car operating either in normal service or according to a fixed program which includes

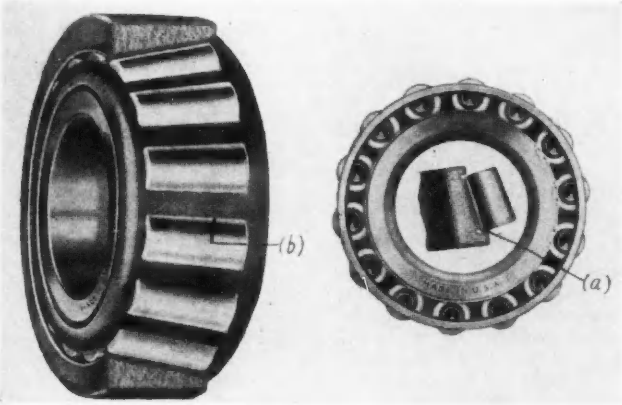


Fig. 20 – Points in Taper-Roller Bearings Subject To Wear When Run In Abrasive Lubricants

various types of load conditions that might be met in service, such as would be found in heavy pulling in sand roads and in high-speed cruising. Such tests give comparative results. Where it is desired to obtain very accurate results on the comparison of different lubricants as to their effect on wear, severe loadings are not required. Dynamometer tests run at loads required to propel the car at 50 m.p.h. on level roads will give a very accurate comparison of lubricants.

In either type of test special care is exercised in the assembly of the gear units to insure that they are as nearly alike as possible, and accurate measurements are made on the gears and bearings and on the positions of the gears, backlash, and so on, so that both the amount of the wear and the place where it occurs can be determined accurately. It seems to me that, under such conditions, an axle which has been run for 50,000 miles and which shows half as much wear as another axle which is run 25,000 miles, and that we can safely say that this is a true indication of the abrasiveness of the lubricants. Fig. 22[5] shows excessive wear on a hypoid rear-axle

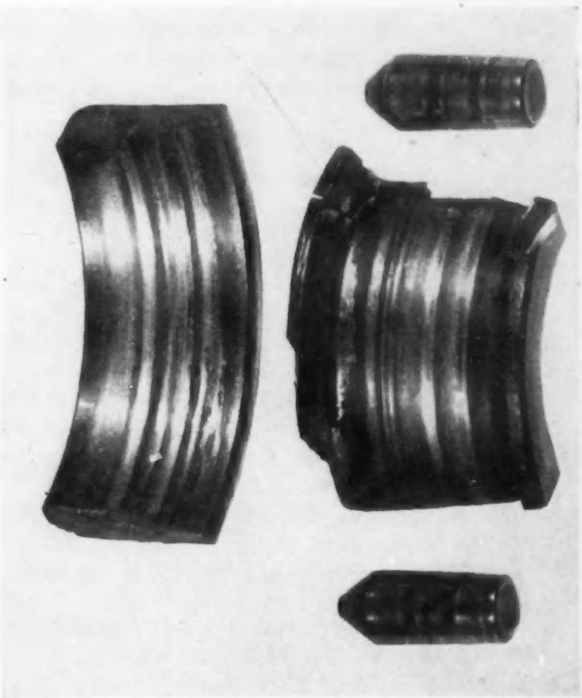


Fig. 21 – Failure of Taper-Roller Bearings Due to Lack of Lubricant

pinion caused by poor lubricant. Also evidence is obtained by examining the lubricant itself at the conclusion of these tests, such as measuring its load-carrying capacity and abrasiveness on the Timken testing machine and by analyzing the oil for iron content and other substances which may indicate wear.

It is a definite fact that some E.P. lubricants are made from mineral oils which contain abrasives at the very outset. We frequently have found at the conclusion of a test, a white solid pasty substance centrifused out on the inner surfaces of the bearing cages and, after careful chemical analysis, we found that it was foreign to any of the materials used in compounding the oil – nothing but clay. Some mineral oils used in the compounding of E.P. oils have been found by chemical analysis to contain clay, which is a highly abrasive substance.

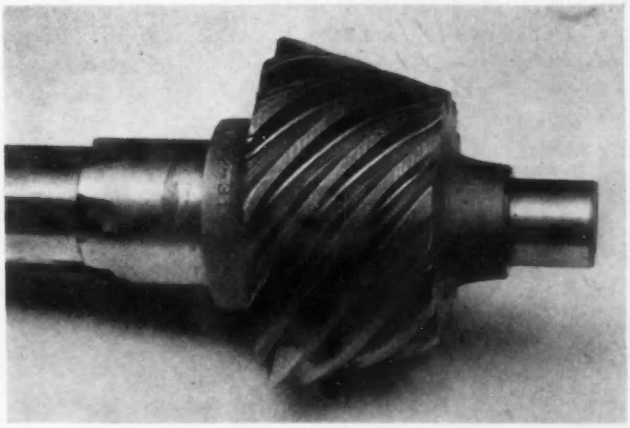


Fig. 22 – Wear on Hypoid Rear-Axle Pinion (Approximately One-Third of Original Tooth Thickness) Occurred In Less Than 300 Miles of Operation on a Mild Sulphur-Saponifiable Lubricant – This Pinion Operated for Approximately 5000 Miles Without Wear on a Powerful Lead-Soap Active-Sulphur Extreme-Pressure Lubricant

[5] "The Lubrication Requirements of Automotive Rear Axles," by H. R. Wolf, Proceedings, Seventeenth Annual Meeting, American Petroleum Institute, Section III, p. 37, Fig. 4.

Oils that are unstable in long-time tests show a considerable diminution of load-carrying capacity and a great increase in abrasiveness as indicated by the iron content in the lubricant.

Summary

In the foregoing part of this paper considerable effort has been made to explain the nature of hypoid gears and to try to convey some idea of why straight mineral oils are not suitable and that special oils are required. Also, I have tried to convey the idea of how much effort is being made by the axle designer and the manufacturer to attain, as nearly as possible, perfection in building the axles. I have tried to give you some idea of what more than 10 years of development work directed exclusively to hypoid axles has yielded. And now, I shall try to convey the idea that all of these "pains" and all of the benefit for this work can become an absolute and total loss by the use of an unsuitable lubricant.

The aim of the car manufacturer, of course, is to have his product operate in the hands of his public with the nearest approach to perfection that is possible. He is not interested in selling oils and, if it were possible, he would have nothing to do with lubricants. However, when he is faced with the condition which now exists, it is necessary for him not only to become vitally interested in lubricants, but to exercise every power at his command to see that his product gets the right kind of lubrication.

The choice of the car lubricant by the car manufacturer will be based on a rating of the five items listed as follows: (1) E.P. value, (2) low abrasiveness, (3) high stability, (4) low chemical activity, and (5) proper flow properties.

And he will use the method at his command for rating these various items but, more than anything else, he will be influenced by their ability to operate in his automobile.

In the choice of a lubricant, mere viscosity has no advantage from a load-capacity standpoint. The common notion that the body of the oil determines the load-carrying capacity has absolutely no value in judging the load capacity of E.P. lubricants. Furthermore, heavy or thick oils do not reduce noise and, consequently, have no value as noise suppressors. What is absolutely important is that the oil must flow readily in the coldest weather and if it becomes thick enough to channel, the axle can be ruined in five minutes.

On the other hand, the oil must not have the characteristic, like some of them do, of becoming stiff when beat up with entrapped air when the axle is run at high speed. Some of these oils become stiff and will channel just like the white of an egg may be beat up and become stiff with the entrapped air. Such oils return to their original state when the air is removed by slight heating and stirring. Some oils simply foam badly, which is definitely injurious as, in some cases, it builds up sufficiently high pressures within the axle, even when vented, to cause seals to leak and, therefore, to lose oil.

Oils that oxidize readily are unsatisfactory. Some oxidize so badly that it is not merely a matter of the viscosity increasing during the course of use, but the oil actually "cokes" in the axle. Such oils cause damage quickly and are totally unsatisfactory.

A word about service is pertinent here because, with a good stable E.P. lubricant, frequent service is not at all necessary. It is, of course, necessary to keep the axle filled to the proper oil level, and it would be desirable simply to add oil occasionally if there were any leaks, but oil changes more frequent than once a year are wholly unnecessary.

Naturally, the element of abrasiveness must be as low as possible and change as little as possible during use. This desired stability, of course, also applies to the E.P. value or load capacity of the lubricant.

One rule that the user of E.P. oils can follow, and in this rule I also am including those who have to do with the lubricating service, is to use the lubricant recommended by the manufacturer of the particular car. I am certain that the car manufacturer has a better chance of knowing more accurately than anyone else what lubricant is most suitable for his axle and, if his recommendations are followed, the best results will be obtained.

Much can be said, of course, to the manufacturers and distributors of hypoid lubricants. I am sure that, if the producers and the distributors will put forth as much effort to build satisfactory lubricants as the car manufacturers have to build fine mechanical units for axles, the problem facing us now will not be concerning the types of lubricants but rather, their individual properties, and the poor attempts to make E.P. lubricants with the resulting poor products will be eliminated. There are some very good hypoid lubricants. There are many who are earnestly endeavoring to produce good lubricants and for these there is no criticism.

Discussion

Foaming Characteristics of Gear Oils

– G. L. Neely

Standard Oil Co. of Calif.

THE company which Mr. Griswold represents has been a pioneer in the use of hypoid gearing and his excellent paper presents some of their fundamental knowledge that is so necessary to produce hypoid axles having long and satisfactory life. The subject of hypoid gear oils also has been covered, and we are waiting with interest to see how the hypoid lubricants perform in the 1937 cars and how well service value can be correlated with laboratory tests, and particularly in regard to film strength as measured in the S.A.E. Tester.

There is one feature of gear lubrication that has not been stressed in Mr. Griswold's paper, nor in any other with which I am acquainted, and that is the subject of foaming. It is well known that leakage of gear lubricants from rear axles to brakes is a serious source of trouble and is largely attributable to foaming. Foaming is also a serious detriment in transmissions, as the lubricant may be forced into the driving compartment. It seems timely to present some data that we have obtained in this regard.

The foaming characteristics of lubricants vary with the type of lubricant and also with the mechanism being considered. The following results from a truck transmission illustrate this point:

Lubricant	Volume Increase, per cent
Lubricant No. 1, Mild E.P. – S.A.E. 160	35
Lubricant No. 2, Mineral Oil *A* – S.A.E. 160	40
Lubricant No. 3, Mineral Oil *B* – S.A.E. 160	30
Lubricant No. 4, Mineral Oil *C* – S.A.E. 250	45

These results show the type of variation to be effected between types of lubricants. Similarly, the performance of the same lubricant may be appreciably affected by the type of gear set being considered. The following laboratory data illustrate this point for Lubricant No. 1.

Gear Set	Volume Increase, per cent
Truck Transmission *A*	35
Truck Transmission *B*	15
Ford Transmission	25
Studebaker Differential	20

With the foregoing values in mind, tests were undertaken to study some of the factors which were thought to have an effect on foaming such as temperature, viscosity, rate of agitation, and so on. It was found in the Sunbeam Mixmaster, for example, which device has been used by

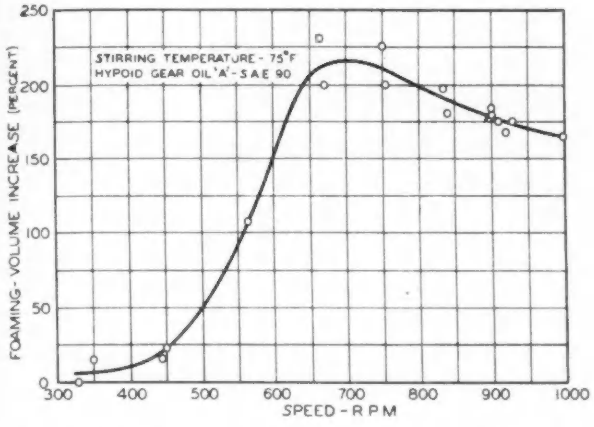


Fig. *A* (Neely Discussion) – Foaming Characteristics of Gear Oils – Effect of Stirring Speed as Measured in Sunbeam Mixmaster

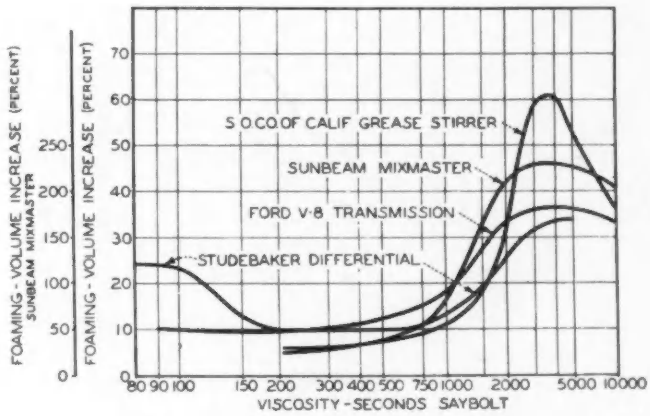


Fig. *C* (Neely Discussion) Foaming Characteristics of Gear Oils in Each of Four Devices

several laboratories for evaluating gear oils in regard to foaming, that the rate of agitation influenced the degree of foaming. This effect is illustrated in Fig. *A*, which shows that the maximum foaming occurred at a speed of 660 r.p.m., or in the No. 4 speed. Similar results were obtained in a Studebaker differential which are as follows:

Speed, r.p.m.	Volume Increase, per cent
420	24
760	29
1200	16

The most striking data in regard to foaming were obtained by changing the temperature and the grade of the oil. The testing devices used were the Sunbeam Mixmaster, a Studebaker differential, a Ford transmission, and a special laboratory stirring device called the "Grease Stirrer." This latter machine consists essentially of a container with an electric heating element for temperature control and a propeller for agitating the oil. The oils used in these tests consisted of the S.A.E. 90, 160, and 250 grades of essentially the same mineral oil and a hypoid-gear lubricant of the S.A.E. 90 grade.

Data obtained for all four oils in all four machines and over a temperature range of 75 to 200 deg. fahr. are shown on the same chart in Fig. *B*. It is seen that the points form a band reaching a maximum at a viscosity of 3000 to 5000 sec. Saybolt. The curves for each of the different devices are shown on Fig. *C*. Attention is called to the marked similarity in the shapes of the curves from the different machines. It will be noted that a smaller scale was used in plotting the results obtained with the Mixmaster.

The real significance of the effect of viscosity on foaming comes in taking foaming measurements. Obviously, if two oils are to be compared, they should be compared over the ranges of temperature to be encountered. Another significant point is that, as a gear unit is usually started up with a cold lubricant, the temperature rise generally causes the lubricant to go through the viscosity range corresponding to maximum foaming.

It is to be understood, of course, that oils differ widely in foaming characteristics and that viscosity, although an important variable, is only one of the characteristics of lubricants influencing foaming. It does appear, however, that, in general, gear oils give their greatest degree of foaming in gear cases when the operating gear-oil temperature is that at which the gear lubricant has a viscosity of 3000 sec. Saybolt.

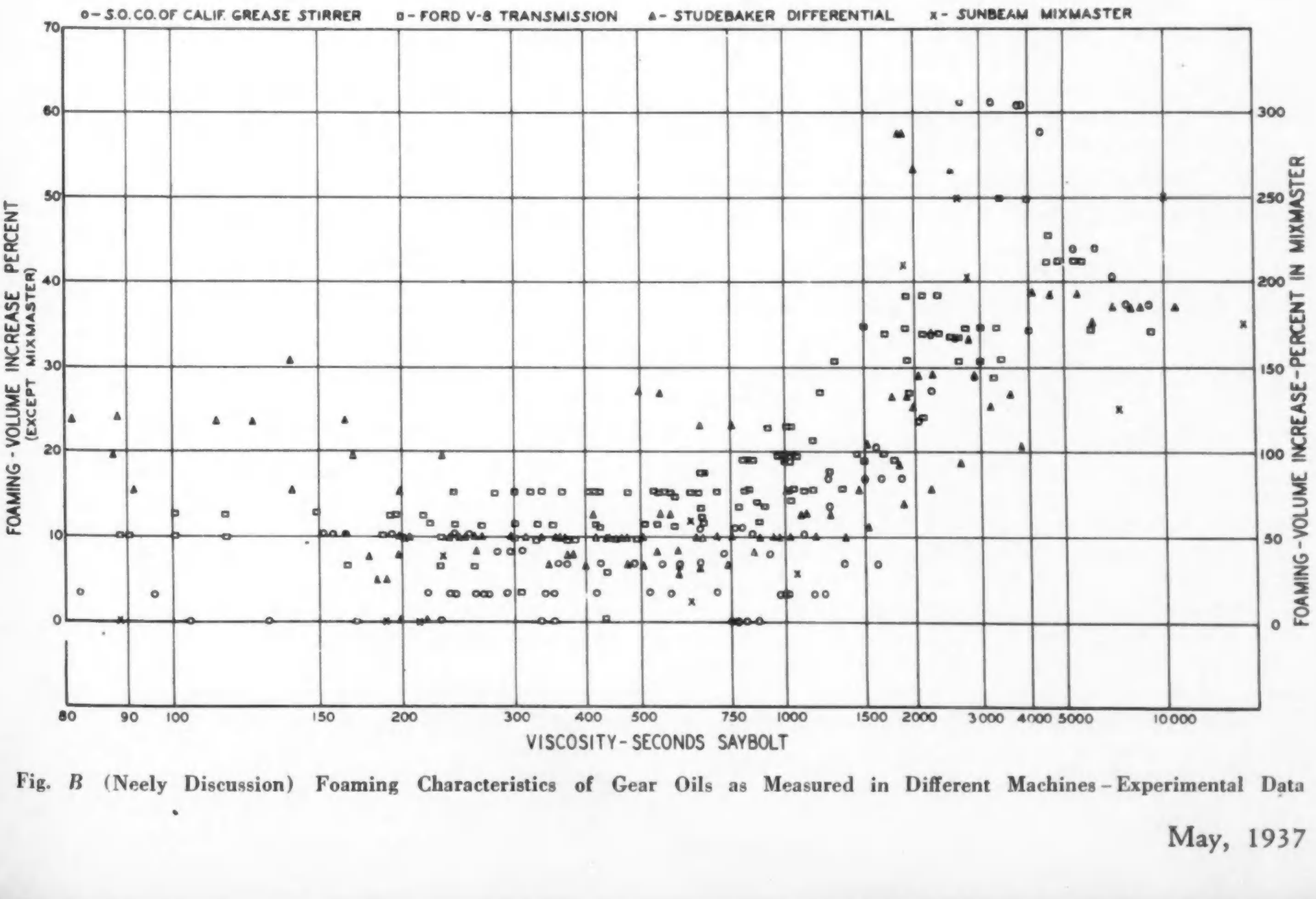


Fig. *B* (Neely Discussion) Foaming Characteristics of Gear Oils as Measured in Different Machines – Experimental Data

Automatic Transmissions

By P. M. Heldt
Engineering Editor, "Automotive Industries"

THIS paper presents an unusually comprehensive review of automatic transmissions, past and present – domestic and foreign. Operating principles, limitations, advantages, and disadvantages of each of many types are explained with the aid of cross-sections and diagrammatic sketches.

The author points out that automatic transmissions are not new, the first one being developed in 1900, and the first car equipped with one was placed on the market in 1904.

This first transmission, invented by George S. Strong, employs a roller-ratchet drive and is of the infinitely-variable type. Also described under this heading are the de Lavaud automatic transmission and the R.v.R. automatic torque converter. Hydraulic types of infinitely-variable automatic transmissions discussed include the Waterbury and the Lysholm-Smith hydraulic torque converters. Inertia-type automatic transmissions take up the Spontan, the Constantinesco, and the Hobbs. In the self-shifting category are grouped the Sturtevant, Yellow Coach, Macallen, Prince and Tyler.

In the review of differential transmissions that concludes the paper are included electric and hydrostatic, the Entz electric transmission, the Electrogear unit, and the Bendix Turbo Flywheel Gear.

IN the early years of the automobile industry when steam cars and electric vehicles furnished a standard of comparison, the control system of the gasoline automobile was generally regarded as crude and complicated. On the steam car a foot-operated throttle gave a continuous, smooth variation of driving torque over the whole range from zero to the maximum, and this operation was considered the ideal; on the electric, although changes in the driving torque took place in perceptible steps, there was no interruption in the drive as the change occurred, and the control, there too, was by a single device – the controller lever.

[This paper was presented at the Philadelphia Section Meeting of the Society, Philadelphia, Pa., Feb. 10, 1937.]

Several different types of transmission were used on the early gasoline cars, including the individual-clutch type, the planetary transmission, the sliding-gear transmission, and the friction drive. A highly flexible drive was badly needed at the time, owing to the lack of flexibility of the early engines, and it is, therefore, little wonder that the idea of an infinitely variable transmission for automobiles occupied the minds of inventors at an early date. Such a mechanism was developed by George S. Strong of New York as early as 1900, and a brief description of his transmission will be given a little farther on.

Aside from their lack of sufficient flexibility, the chief objection to the early transmissions was that they were difficult to handle. It was believed that this difficulty could be overcome by making the gear change or shift automatic, and it is a noteworthy fact that the first car with an automatic transmission was placed on the market as far back as 1904. This was the Sturtevant car produced in Boston. An illustrated description of its transmission appeared in *The Horseless Age* for Aug. 10, 1904.

By 1910 the sliding-gear transmission had come into use on nearly all stock cars, the one important exception being the Ford Model T, which retained the planetary gear until 1928. There was no basic change in transmissions during the second and third decades of the present century, although a number of cars were equipped with so-called pre-selective transmissions, with which, by means of a small lever carried on the steering post, the driver at any time could set the mechanism for the change he expected to become necessary next, and then, when the time for the change arrived, he would press down on the clutch pedal (or let up on the accelerator pedal) and the gear would be changed in accordance with the setting of the preselector lever – either by physical force exerted on the clutch pedal, by a solenoid, or by a vacuum cylinder. The so-called Electric Hand in use today belongs to this type of control mechanisms.

Toward the end of the 'twenties the Warner Gear Co. brought out its Hi-Flex transmission, a four-speed design incorporating internal gears, and although this design did not prove a permanent success, it caused such a stir in transmission circles that it must be considered as definitely having reopened the transmission problem. Since that time transmission gears have been made silent, and the gear-synchronizing device was introduced, which practically eliminated clashing and made shifting possible under all conditions, with the result that the conventional sliding or shifting transmission has become a

rather satisfactory device – light and compact, highly efficient, easy to operate, silent in operation, and inexpensive to manufacture. Of course, a continuously-variable type would be preferable.

There seems to be only one really serious objection to the present type of conventional transmission, and that is that it calls for the use of three pedals for the control of the car. Two of these pedals, the accelerator and the brake, must be operated by the same foot, and the necessary shifting of the foot from the accelerator to the brake pedal adds to the time lag in bringing the car to a stop in an emergency; besides, it is inconvenient and, in addition, there is the possibility of the foot slipping from the brake onto the accelerator pedal, or of the accelerator being depressed instead of the brake pedal by mistake. It would seem, therefore, that a transmission which reduces the number of pedals required to two would be a real improvement.

As hinted in the foregoing, transmissions may be divided into stepped and continuously-variable types. Either of these types, of course, may be controllable or automatic. The driver in every case has control over the speed of the car by means of the accelerator. With some stepped transmissions generally called "full automatic," the ratio of crankshaft revolutions to propeller-shaft revolutions (the gear ratio) is always the same for any given car speed or engine speed, which involves that the change from one ratio to a higher or a lower one always occurs at the same car speed, leaving the driver no control. It is now generally admitted that this arrangement is undesirable and that the driver should have a choice of ratios at any point of the speed range. Transmissions with which this choice is possible and in which changes in ratio normally occur automatically are referred to as semi-automatic.

Infinitely-Variable Transmissions

The first type of automobile transmission which made it possible to have an infinite number of gradations in the transmission ratio was the friction-disc drive. This drive was regular equipment on some four or five early cars, but it was only moderately successful with the low engine powers of that period, and it would be entirely impractical with our modern passenger-car engines of around 100 hp.

Another method of obtaining infinite variations in the speed ratio between driving and driven shafts is by the use of roller ratchets, which also are known as "mechanical valves." This mechanism can be made to transmit motion in both directions, so combining a reverse gear with a variable-speed gear for forward drive. Such a reversible roller-ratchet drive was invented by George S. Strong of New York, who for a long time was prominently connected with the roller-bearing industry.

A sectional view of Strong's reversible roller ratchet is shown in Fig. 1. It will be seen that a disc with a polygonal outer surface is keyed to the driven shaft. This disc is surrounded by a roller cage containing one roller for each of the "flats" on the disc. The cage and rollers are surrounded by an outer race of hardened steel. The roller cage is connected to the steel center by means of two straight keys, each of which has a helical feather key on its outside, so that by moving the key lengthwise in its seat, the angular relation of the roller cage to the steel center can be changed. Normally the rollers are located at the center of the flats or cam surfaces of the steel center but, if the keys are moved in one direction or the other, the rollers approach the edge of the cam surface and, if the outer rings or races to which the connecting rods are attached are then moved angularly, the rollers are wedged in between the steel discs and the outer races, and angular motion is transmitted from the outer race to the steel center. The angular movement imparted to the steel center depends on the stroke or "throw" of the connecting rods, and means are provided by which the stroke can be varied gradually from zero to the maximum.

In the Strong transmission, which was fitted to a motor truck known as the Union built in Philadelphia, the connecting rods connected to a crankpin projecting from the face of the engine flywheel. The crankpin was secured to a piston located in a radial cylinder machined in the flywheel; it was forced toward the center of rotation by a coil spring and away from it by oil under pressure which entered the cylinder through a drilled passage extending the whole length of the crankshaft. The driver controlled the "gear ratio" by starting and stopping the oil pump, which was driven from the engine.

In 1925, Sensaud de Lavaud, who had achieved fame and fortune through the invention of a process of casting iron pipes centrifugally, came forward with an automatic transmission for automobiles, of which a sectional view is shown in Fig. 2. It embodied a number of roller ratchets on the rear axle, actuated by connecting rods extending to them from a wabbleplate mounted on the driving shaft. The angularity of the wabbleplate, and therefore the stroke of the connecting rods, was variable, and it varied automatically in accordance with the torque load on the axle, that is, with the resistance to motion. A spring consisting of a series of cupped washers tended to increase the angularity of the wabbleplate, whereas the torque reaction tended to decrease it. Assuming that the car has been running along under certain fixed operating conditions and that the wabbleplate has been held in a definite position by the balance between the spring pressure and torque reaction, if the resistance to the motion of the vehicle increases, the spring will be compressed farther and the angularity of the wabbleplate will be decreased, so that the stroke of the connecting rods is reduced, the car speed is reduced, and the engine is able to produce the greater driving torque necessary to overcome the greater resistance to motion.

With a transmission of this type it is possible to keep the engine operating under practically full torque regardless of the speed of the car, from which considerable gain in economy may be expected. De Lavaud applied his transmission to a number of cars and claimed a decrease in the fuel consumption of between 15 and 26 per cent, and a decrease in the oil consumption of as high as 50 per cent. This latter figure, although quite

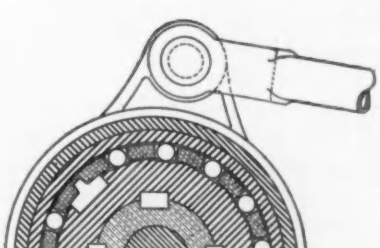


Fig. 1 – Strong's Reversible Roller Ratchet

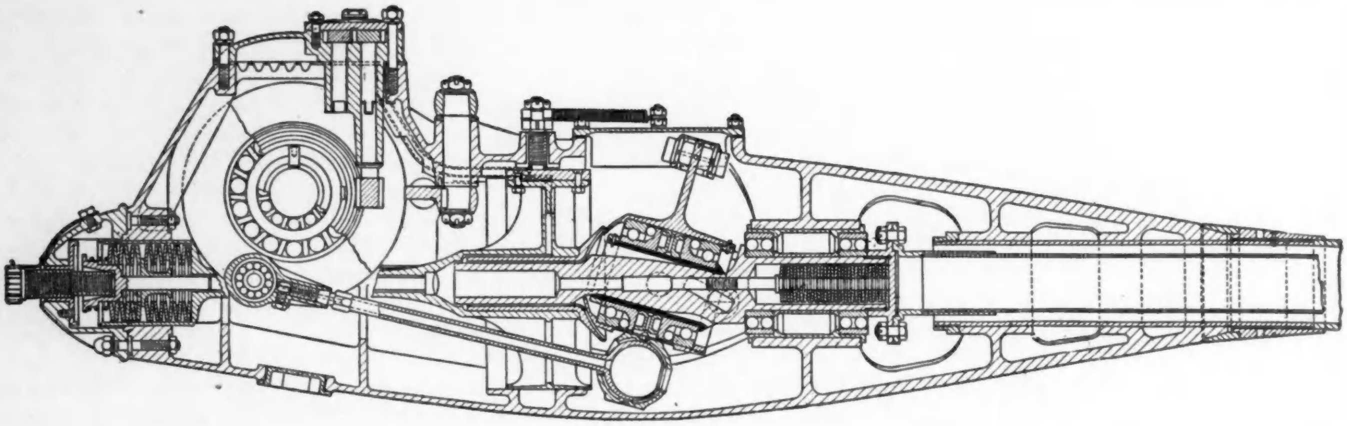


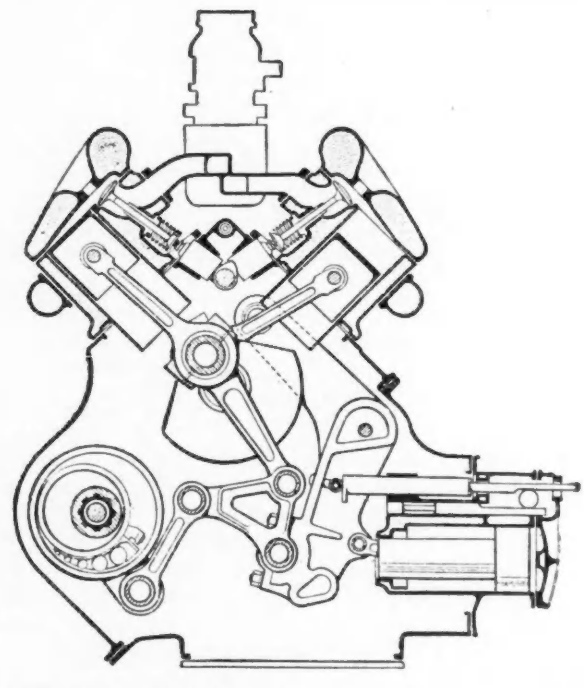
Fig. 2 – De Lavaud Automatic Transmission, Wabbleplate and Roller-Ratchet Type

high, is credible in view of the known fact that the rate of oil consumption of an engine varies rapidly with changes in speed, and an infinitely-variable gear can be taken advantage of to hold down the engine speed. The economy figures given are based on a comparative test of two Voisin 10-hp. cars, one with the conventional transmission, the other with the de Lavaud drive, both cars being driven over the same route. Among the disadvantages of this transmission brought out by the test were that the rear-axle unsprung weight was increased by about 25 per cent and that the wabbleplate was unbalanced and set up resonance in the axle tube. In recent years de Lavaud has given up the wabbleplate type and has been working on hydraulic transmissions.

The latest development in the line of "variable-throw" automatic transmissions is the R.v.R. torque converter, which is regular equipment of the Minerva front-drive car exhibited at the recent Brussels show. This car has a 90-deg., V-8 engine mounted transversely in front and built integral with the transmission and differential housings. Independent suspension is used and the differential housing, therefore, is carried on the springs. The final drive is accomplished through four roller ratchets whose impulses are equally spaced. As may be seen from Fig. 3 (reproduced from *The Autocar*), the outer member of each roller ratchet connects by a link to a triangular rocking member which, in turn, is linked to one of the throws of the engine crankshaft. The triangular member is pivoted to a transverse shaft held in a swinging frame, the frame being pivoted on an axis above and behind the crankshaft axis.

With the swinging frame in the position shown in Fig. 3, the link connecting to the roller ratchet is in a substantially tangential position, and the angular motion of the ratchet per cycle is therefore a maximum. For this setting the reduction ratio between crankshaft and differential gear is 2:1. It can be seen readily that, if the pivot for the triangular member is moved farther away from the roller ratchet, the former and its link connection to the roller ratchet come more nearly into line, and a given oscillating motion of the triangular member then produces a smaller angular motion of the arm on the roller ratchet. In fact, with the pivoted frame in the extreme position to the right, oscillations of the triangular member produce no motion of the ratchet arm, so that the engine can be running without moving the car. Any intermediate speed ratio can be obtained and the gear is therefore continuously variable between the limits of 2:1 and infinity.

The position of the swinging arm is controlled by a stepped piston located in a double-ended cylinder, whose two ends communicate with the engine lubricating system. The piston can be subjected to the oil pressure in either direction, and flow of oil to one end or the other is controlled by a valve which is interconnected with the accelerator pedal. Unfortunately no details with respect to this control mechanism are available.



Courtesy The Autocar

Fig. 3 – R.v.R. Automatic Torque Converter

Hydraulic Transmissions

Another type of infinitely-variable transmission on which a great deal of work has been done with a view to adapting it to automotive uses is the hydraulic. Hydraulic transmissions naturally divide into two classes, hydrostatic and hydrodynamic. In the former the working fluid is placed under pressure in, and set in motion by, a pump, usually of the multicylinder plunger type, and the fluid moved by this pump acts on the

pistons of a hydraulic motor. The rate at which power is being transmitted by such a device is measured by the product of the fluid pressure by the volume displaced by the plungers of the pump in unit time. In order to change the transmission ratio, the stroke of one of the two elements is varied, usually that of the pump. For instance, if the pump stroke is reduced to one-half, the speed of rotation of the pump and the horsepower input remaining the same, then the pressure to which the fluid is subjected will be doubled, and with twice the fluid pressure the torque on the shaft of the hydraulic motor will be doubled, while its speed will be halved, because of the lower rate of delivery by the pump.

One of the pioneers in hydrostatic transmissions was Charles M. Manly, a former president of the S.A.E. Hydraulic transmissions for other than automotive purposes are being manufactured by the Waterbury Tool Co., Waterbury, Conn., and a sectional view of this transmission is shown in Fig. 4. It comprises a pump unit and a motor unit, arranged end to end, with a valve plate in between, all in the same housing. The pump unit is shown at the right and the motor unit at the left. Both units are of the round or barrel type, the plungers connecting the swashplates by means of short connecting rods. The swashplate of the pump unit is mounted in a tilting box whose inclination can be varied from zero to the maximum by means of a hand control.

It seems that with the great increase in the power of passenger-car engines during the past two decades, such hydrostatic transmissions have become entirely unsuitable for passenger-car use. All of the power transmitted is at all times converted first into hydraulic power and then converted back into mechanical power, and the double conversion involves considerable losses. It is rather doubtful whether such a transmission in automobile use under normal conditions would show more than 75 per cent efficiency, and a loss of 25 per cent of the power in the transmission would be highly objectionable. Moreover, if under hard driving conditions some 20 hp. were wasted in the transmission, it would be difficult to keep the oil in the units reasonably cool.

In hydrodynamic transmissions a fluid is set in motion by an impeller provided with blades or shovels. In this case power is being transmitted by setting the fluid in motion by the driving member and then letting it spend its kinetic energy on the blades of the driven member.

If both driving and driven member are inclosed in the same housing and the fluid passes directly from the blades of the driving to those of the driven member, the torque on the driven member will be no greater than that on the driving member, and as the speed of the driven member can never quite equal that of the driving member, we have a device akin to a slipping clutch. This device is widely used in England, and to some extent in France, in combination with planetary transmissions, being generally referred to as a fluid flywheel. In more correct technical terminology it is known as a hydraulic coupling. The various speeds of the planetary gear are engaged by applying friction brakes to drums associated therewith. There is no shifting of gears or of dog clutches engaging gears, and the fluid flywheel introduces a certain flexibility which prevents shocks. Although there is a slight loss in the "flywheel," under normal driving conditions it amounts to only a few per cent, and it is, therefore, much more efficient than the hydrostatic type of transmission. Of course, it must not be understood that the fluid flywheel is a substitute for, or an equivalent of, the hydrostatic transmission; it is not, since it is incapable of multiplying the torque.

The reason the fluid flywheel cannot increase the engine torque is that it has no member on which any additional torque could react. To obtain an increase in torque, a third set of blades must be provided, carried on a member that is mounted rigidly on the engine or on the chassis frame.

Hydraulic couplings and hydraulic torque converters can be combined with mechanical units in various ways to secure automatic changes of gear ratio together with high efficiency of operation under most driving conditions. A transmission embodying this feature is the Lysholm-Smith, which was developed originally in Sweden and is now being manufactured in England by the firm of Leyland Motors, Ltd., and in Germany by the Krupp Works, for both bus and railcar installations. All multiplication of engine torque for acceleration and hill climbing is effected hydraulically, and changes in "gear ratio" take place automatically. When the vehicle ap-

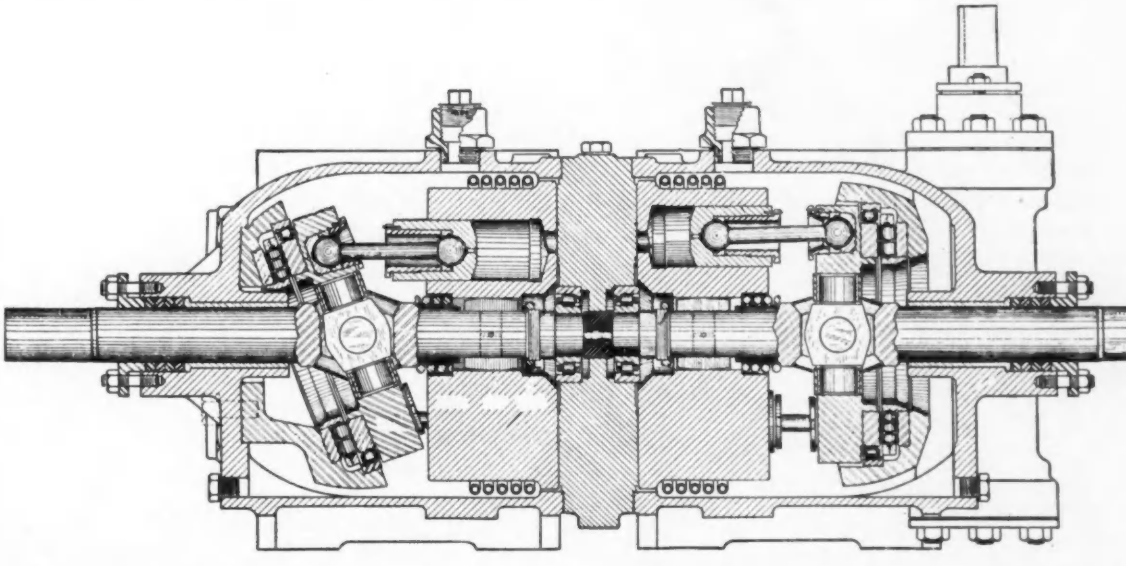


Fig. 4 – Waterbury Hydraulic Torque Converter (Hydrostatic Type)

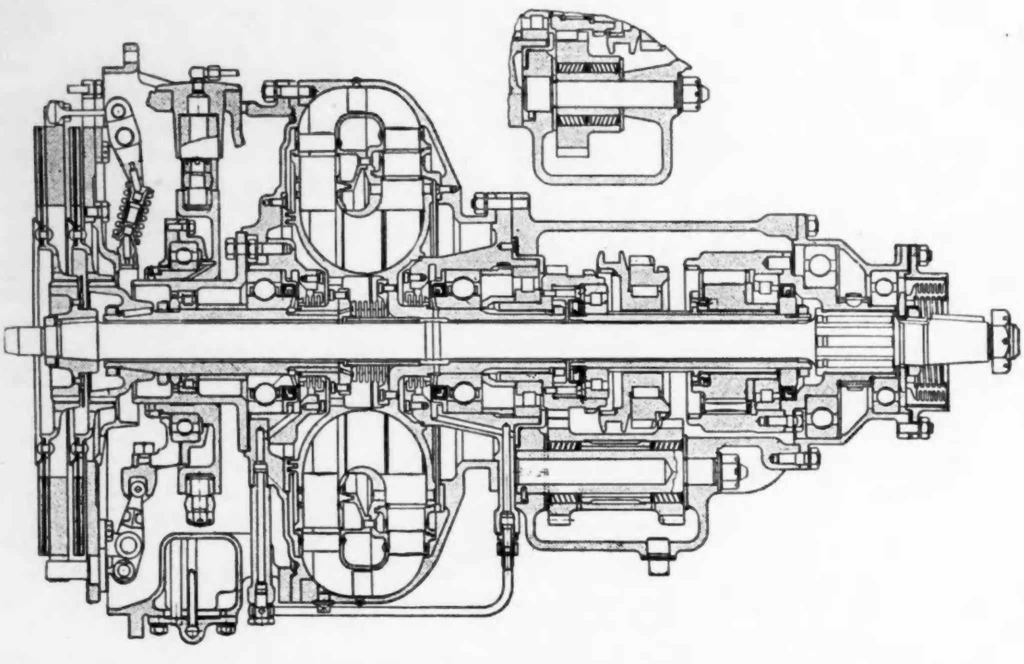


Fig. 5 – Lysholm-Smith Hydraulic Torque Converter with Direct-Drive Feature

proaches the normal driving speed, the driver sets the direct-drive pickup lever by hand, and thereafter the drive is direct. Reverse is obtained by means of a special reversing gear which is combined with a double friction clutch and the hydraulic mechanism. There are four control members, namely, the accelerator pedal, the brake pedal, the direct-drive pickup lever, and the reversing lever. The last is normally in the position for forward drive and needs to be moved only when it is desired to back up. For railcars these transmissions are built without reverse.

Referring to the sectional view of this transmission (Fig. 5), at the left is seen the dual friction clutch, which is shown in the direct-drive position, the crankshaft being coupled directly to the propeller shaft. With the clutch control lever in the opposite position, the crankshaft is connected to the tubular shaft which carries the impeller of the hydraulic torque converter. It is not necessary to accelerate the vehicle with the friction clutch; this clutch can be engaged fully before the vehicle starts, and acceleration is accomplished by means of the hydraulic unit, by opening the throttle and speeding up the engine. When the direct-drive pickup lever is engaged, the vehicle is already traveling at a speed close to normal, and there is, therefore, a minimum of slippage and a minimum of wear and tear on the clutch linings. This change, moreover, is made without interruption in the torque.

The hydraulic torque converter comprises a centrifugal pump and a three-stage hydraulic turbine in the same housing. The turbine wheel is connected to the propeller shaft through a free-wheeling unit, whose object is to disconnect the hydraulic unit from the drive mechanism when the direct drive is used, so that the impeller and turbine wheel will not rotate and will not cause any hydraulic losses. When making the change to direct drive, the operator lets up on the accelerator pedal slightly.

A number of transmissions have been developed in which the inertia or centrifugal force of moving weights is made use of to transmit power from one shaft to another and to vary the ratio of the torques on the two shafts. In a number of these transmissions use is made of masses which rotate with the flywheel of the engine but are so mounted on it that they can move in and out from the axis of rotation. When these masses are in the inner position, close to the axis of rotation, they have a certain amount of potential energy, the same as a mass which has been raised a certain distance above ground level, and in moving from the inner to the outer position they may be made to do useful work, such as turning the propeller shaft of a car. While the masses are brought back to their inner position, they absorb an equivalent amount of energy, which must, of necessity, come from the engine. It is therefore necessary that during each cycle the moving masses be placed in driving relation first with the engine, to bring them into their inner position, nearest the axis of rotation, and then with the driven shaft, in order to utilize the energy stored up in them for the propulsion of the car. Some sort of rapid-acting clutch is therefore necessary, and the device used is a development of the roller ratchet described in an earlier paragraph.

One such transmission was developed in Sweden by Dr. F. Ljungstrom and was marketed as the "Spontan." A diagram of the principal parts of the mechanism is shown in Fig. 6. Two "bobweights" are mounted pivotally on the flywheel and rotate with it. These bobweights are connected to eccentric straps surrounding eccentrics on the driven shaft. The centrifugal force on the bobweights tends to make them move radially outward, and through the eccentrics they exert a turning moment or torque on the driven shaft; if the resisting

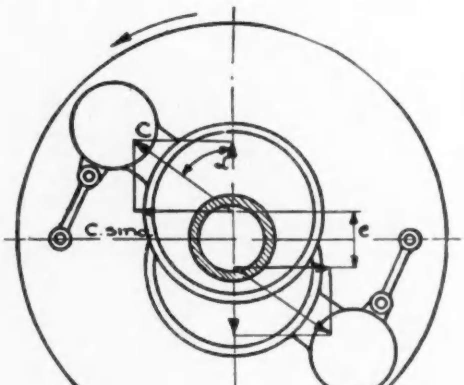


Fig. 6 – Diagram Showing Inertia Weights and Eccentrics of Spontan Automatic Transmission

moment on this shaft is less than the driving moment, the shaft will be turned.

As long as the car is at rest, the eccentrics are stationary and, if the engine is idling, the bobweights will be turning and at the same time they will be reciprocated between their extreme positions with relation to the axis of the shaft. At a constant engine speed the bobweights will then impress an alternating torque on the eccentrics, which may be represented by a sine curve, the same as an alternating current. The direction of the torque changes twice during every revolution of the flywheel. These alternating torque impulses must be "rectified" before they can be utilized for the propulsion of the vehicle, and this requirement is accomplished by means of the double roller-ratchet clutch of which a diagram is shown in Fig. 7. Sections of three concentric sleeves are shown. The middle one is the main driving sleeve on which the two eccentric sheaves are mounted; the inner one is the driven sleeve fastened to the propeller shaft; and the outer one is a reaction sleeve on which the reverse torque is taken up. The two roller clutches between the three sleeves are so arranged that they will be engaged by opposite motions of the main driving sleeve, respectively, which sleeve has an oscillating motion. When moving in one direction its torque is transferred to the driven sleeve on the propeller shaft while, when moving in the opposite direction the torque is impressed on the reaction sleeve. The reaction sleeve is held in position by springs and these springs are tensioned when the reaction sleeve is subjected to torque

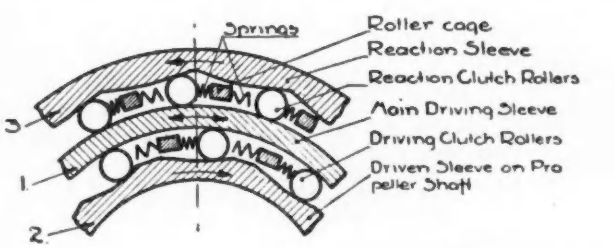


Fig. 7 – Double Roller Ratchet of Spontan Transmission

by the driving sleeve, and they give out the energy so stored during the other half of the cycle.

With a device as here described, the torque on the driving shaft when starting the car is naturally quite non-uniform and, as a matter of fact, the start is very jerky. Once the car is started, the driving torque is evened out by a flywheel on the driven sleeve, and also by the torsional flexibility of a long propeller shaft which winds and unwinds as the torque impulses wax and wane.

It seems apropos here to lay emphasis on the need for a reaction member in any kind of transmission in which the torque is to be multiplied under certain conditions. Inventors sometimes overlook this requirement, and I have been asked repeatedly to give an opinion on transmissions which were enclosed completely in a housing revolving with the flywheel and therefore had no point of reaction on the engine frame or car frame. The principle that action and reaction are equal and opposite applies to torques or moments as well as to forces. The reaction to the engine torque is represented by the side thrusts of the pistons against the cylinder walls. If the speed is reduced in the transmission, the torque should be increased, and any additional torque produced in the transmission must have a point of reaction somewhere.

The Spontan transmission was demonstrated to American

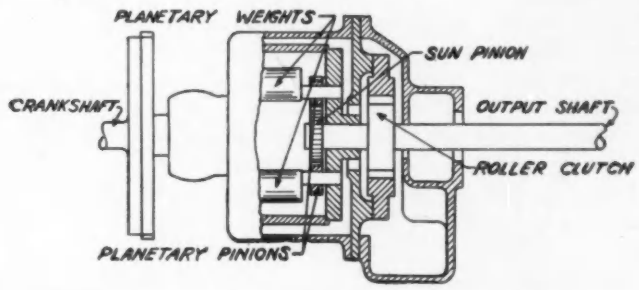


Fig. 8 – Diagram of Hobbs Inertia-Type Transmission

manufacturers in 1930, and the company sponsoring it maintained an office in New York for a time, but it is my understanding that it has not been placed in production on a quantity basis.

Another infinitely-variable transmission in which use was made of inertia masses was developed in England in 1926 by Constantinesco, who made a name for himself by his development, during the period of the World War, of a hydraulic synchronizing gear for machine guns firing between the blades of airplane propellers. Constantinesco built one of these transmissions and applied it to a very small car fitted with a two-stroke engine of only about 30 cu. in. piston displacement.

A somewhat simplified form of inertia transmission, illustrated diagrammatically in Fig. 8, was exhibited at last year's commercial-vehicle show in London. It is known as the Hobbs infinitely-variable transmission and was developed in Australia, or at least was an Australian invention. Connected to the engine flywheel is a drum which contains a sun pinion and a pair of planetary pinions meshing with it. If the sun pinion is held stationary and the housing rotates with the flywheel, the planetary pinions will revolve around their shafts, which latter are carried by the housing. These shafts are provided with unbalanced weights, and when they revolve, the weights naturally tend to assume a radially-outward position. While they are being brought to the inner position, they absorb energy from the engine, and during the next half of the cycle while they move outward again, they are capable of giving out this energy. Back of the sun wheel there is a roller ratchet which transmits only the forward impulses, and the irregularities in the torque impressed by it on the propeller shaft are smoothed out by the torsional flexibility of the latter, which is made of relatively small diameter and quite long. A flywheel at the far end of this shaft also helps to even out the driving torque.

Considering the great technical talent and the large financial resources of some of the men who have devoted themselves to the development of this type of transmission, and the very meager commercial results achieved by them to date, it does not appear that this principle holds out any great promise as to profitable applications in the automobile field.

Self-Shifting Transmission

We now come to the stepped automatic transmission, with either three or four definite ratios, in which the change from one ratio to another occurs either entirely automatically or is brought about by the operator by acting on the speed of the engine (by means of the accelerator) or on the speed of the car (by means of the brake). There are two devices of which quite extensive use is made in such transmissions. One is the automatic friction clutch, the other the overrunning or roller clutch.

There are two general types of automatic friction clutch, one being actuated by the centrifugal force on pivoted masses

revolving with the flywheel, the other by the vacuum or suction in the inlet manifold. The conventional automobile friction clutch is normally held in engagement by springs, whereas the centrifugally-actuated automatic clutch is normally out of engagement. In the latter a number of centrifugal masses are arranged around the clutch, in the form of bellcranks. The pressure plate of the clutch is withdrawn by a spring, and when the engine is idling, the centrifugal force on the masses is insufficient to overcome the pressure of this spring. As the engine is speeded up, the centrifugal force increases with the square of the speed and, at a speed slightly above idling, it overcomes the spring force and the clutch engages.

The vacuum-actuated automatic clutch is normally engaged like the conventional foot-actuated clutch. When the engine is idled, the high vacuum in the inlet manifold, acting on a diaphragm or piston, withdraws the clutch against the force of its springs, but when the throttle is opened the vacuum in the inlet manifold decreases and the clutch is engaged again. To unclutch, all that is necessary is to remove the foot from the accelerator pedal. Vacuum-controlled clutches have been used to a certain extent in conjunction with conventional transmissions for the reason that they make it unnecessary to unclutch by pedal before shifting gears; they are incorporated also in at least one automatic transmission, but with the latter type the centrifugally-actuated clutch is more extensively used.

The Sturtevant, which was the pioneer of this class of "self-shifting" transmissions, in its earliest and simplest form consisted of a train of four gears, arranged in substantially the same way as the first-reduction and intermediate-speed gears of a conventional transmission, combined with two automatic (centrifugal) clutches. When the engine was speeded up, one of the clutches, connected to the driving pinion of the transmission, would engage, and the car was then driven in low gear, that is, through the train of four gears. At a certain higher engine speed the second friction clutch engaged automatically. The driven member of this clutch was fastened to the main drive shaft, which extended entirely through the transmission, and the drive was then direct, this being made possible by providing an overrunning clutch in the gear meshing with the drive pinion. This car did not remain on the market very long but, whether its insuccess was due to faults in the transmission or to some other cause, I have not been able to learn.

A semi-automatic transmission is being used on a considerable number of buses built by General Motors Truck & Coach Division, Yellow Truck & Coach Mfg. Co., and in service in New York and Chicago. An order for 225 such buses was placed by a large operating company, and a service record of more than 2,000,000 miles has been piled up with them to date. This transmission, of which a longitudinal section is shown in Fig. 9, is of the so-called all-spur planetary type but, instead of making use of a band brake to lock one member of the planetary assembly to the housing to afford a point of reaction for the additional torque produced, it makes use of a roller clutch for the purpose. This arrangement prevents rotation of the part affected in one direction only, leaving it free to rotate in the opposite direction.

The transmission is not entirely automatic but is set for forward and reverse motion before starting the vehicle, while changes of gear are controlled by the accelerator pedal. By means of a lever located convenient to the operator, the sliding clutch member *A* can be shifted into any of three different positions, "Forward," "Neutral," and "Reverse." In the drawing this clutch member *A* is shown in the neutral position, its teeth engaging solely with teeth on a hub projecting from a bulkhead of the transmission housing. For forward drive, clutch member *A* is shifted to the right so it will engage also with the teeth on the outer member of the roller clutch *B*, thereby locking this member to the housing. For reverse drive, clutch member *A* is shifted to the left. It then engages with the clutch member *C* which is splined to a hollow shaft formed integral with the pinion *D* of the planetary assembly.

The transmission proper receives its power from the engine through an automatic (centrifugally-actuated) friction clutch. In the original design this was of the multiple-disc type, but in the transmission on Yellow coaches it is of the expanding-shoe type, the shoes being brought into engagement with the inner surface of the clutch drum by the action of centrifugal masses. Only the drum, *E,* of this clutch is shown in Fig. 9. From clutch drum *E* the power passes through the overrunning clutch *F* to the input shaft of the transmission, which has the gear *H* formed integral with it. Clutch member *A* being assumed to be in the "Forward" position, the planetary carrier *G* is held from rotation by the roller clutch *B,* as already explained, and power is transmitted through the gear train *H-I-J-K,* the last of these gears being integral with the output shaft *L.* This is the first forward speed, which is engaged by merely speeding up the engine.

It will be noticed that the planetary gearset has three pinions forming a single unit. The

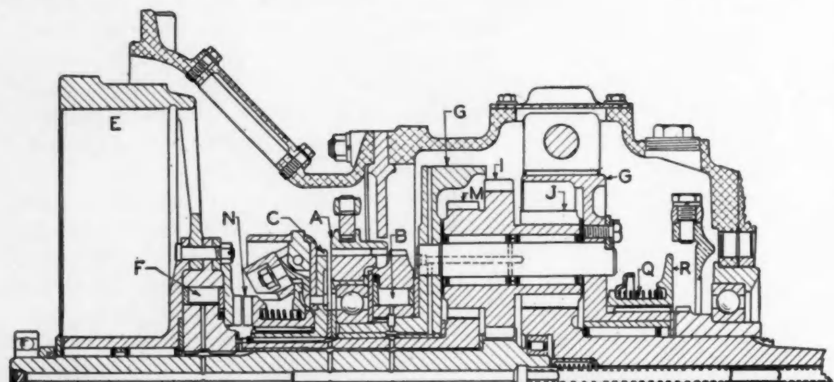


Fig. 9 – Longitudinal Section of Yellow Coach Automatic Shifting Transmission (Monodrive Patents)

third pinion, *M*, meshes with gear *D* formed integral with a tubular shaft, which at its forward end carries a second automatic clutch *N*, of the jaw type. When the car is being driven in first gear the driven or rearward member of clutch *N* turns at a slower rate than the driving or forward member. After the car has attained a certain speed in low gear, the driver momentarily releases the accelerator pedal; this slows down the engine, and when the forward member of clutch *N* has slowed down to the same speed as the rear member, the clutch engages automatically. The drive is now through the train *D-M-J-K*. This makes the planetary pinions turn faster than they would be turned by pinion *H*, but the roller clutch *F* allows pinion *H* and its shaft to free-wheel.

With the transmission in second gear, if the accelerator is momentarily released, the engine slows down, gear *K* becomes the driver, and planetary carrier *G* is speeded up. This action causes the centrifugal masses *O* to fly outward and, through the pins *P* and spring *Q*, shift the clutch member *R* to the right, thereby locking the planetary carrier *G* to the output shaft *L*. The gear is now in direct drive, the whole assembly rotating as a unit. If it is desired to shift to second from high without losing speed, this can be done by disengaging the planetary carrier *G* from the output shaft *L* manually by means of the throw-out shoe *S*. When this operation is done, the transmission is in second gear again.

The reverse is also engaged manually. When sliding clutch *A* is engaged with clutch member *C*, gear *D* and its hollow shaft are held from rotation. Engagement of clutch member *C* is accompanied by disengagement of the outer member of roller clutch *B* by sliding clutch *A*, and the planetary carrier *G* therefore is now free to rotate in both directions. If the engine is now speeded up and the automatic clutch *E* takes hold, pinion *M* rolls around gear *D*, and driven gear *K*, which has a larger number of teeth than gear *D*, will revolve in the opposite direction to driving gear *H*, which has a smaller number of teeth than gear *D*. In this way the vehicle is moved in the backward direction.

This transmission was developed under patents issued to Oscar H. Banker of Chicago. Sole rights under these patents for coach applications were acquired by General Motors Truck & Coach, and all other rights by the Borg-Warner Corp.

In the Macallen transmission, which was developed in Boston, the arrangement of the gears is substantially the same as in a conventional transmission (four pairs on parallel shafts), but three speeds forward are obtained semi-automatically by means of overrunning clutches. There are four of these overrunning clutches in the transmission, one being associated with each of the three forward speeds, while the fourth, located at the rear end, acts as a synchronizer for the direct drive and also makes it possible to use the engine as a brake when in direct drive. The overrunning clutches associated with the direct drive and the intermediate gear differ somewhat from the conventional and are referred to as torque-balancing mechanisms. A sectional view of one of these is shown in Fig. 10. The torque-balancing device is located between two adjacent gears on the same shaft. A lateral flange on one gear forms the outer race for the rollers, which latter surround a cam disc that is free on the shaft, and the spacers between the rollers form part of the second gear. A coiled spring tends to turn the cam disc in such a direction relative to the driven gear with the roller spacers or cage members, that the rollers are held in the disengaged position. But when the torque is reversed so that the car tends to drive the engine, the driven gear with its cage members moves the rollers into the position of engagement, and the drive is then from the first of the pair of adjacent gears through the rollers to the second.

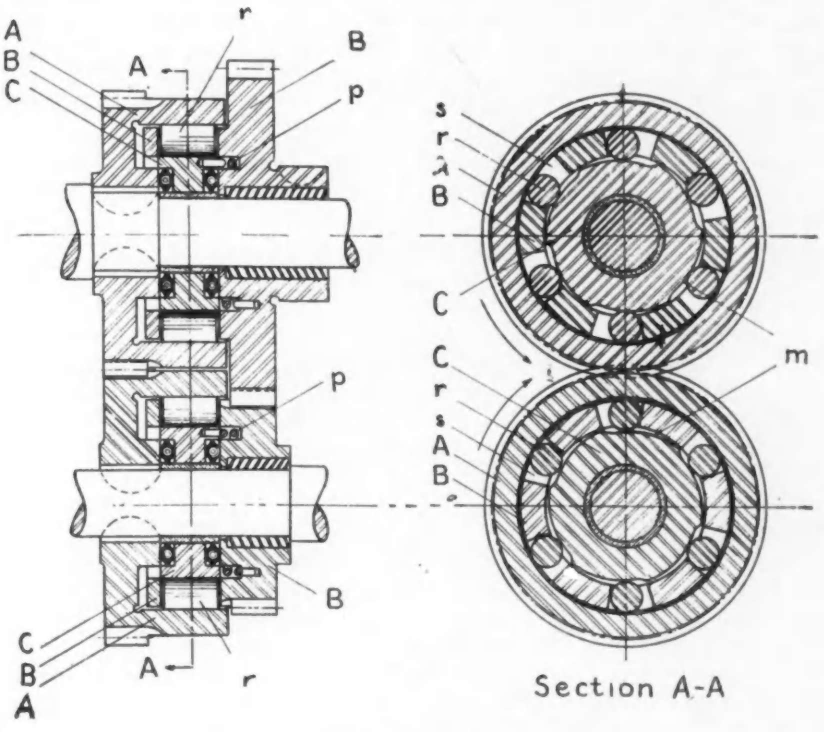


Fig. 10 – Macallen Torque-Balancing Mechanism

The transmission has one shifting member, the movable part of a double positive clutch, which can be set in any one of three positions – Forward, Reverse, and Neutral. Let us suppose that the driver, while pressing down on the clutch pedal, has set the transmission for forward drive. He now releases the clutch pedal and the car starts slowly in low gear, which is through the gear train *3-4-7-8* (Fig. 11), and through the overrunning clutch *9* in gear *8*, and positive clutch *19* to sleeve *11*. After the car has been accelerated in low gear up to a certain speed, the throttle is closed to let the engine idle, and is then opened again. This operation makes the rollers of the torque-balancing mechanism *W* take hold, and the drive is then through the gear train *3-4-5-6* and through clutch member *9* to sleeve *11*. To make the change to direct drive, the throttle is practically closed to bring the speed of the clutch shaft down to that of the propeller shaft. Then the synchronizing clutch at the rear end of sleeve *11* takes hold, locking the sleeve to shaft *1* and causing them to revolve as a unit. Then, when the throttle is opened again, the torque-balancing mechanism *Y* engages and the drive is direct. The driven member of mechanism *W* now overruns its driving member. To change to a lower gear, the driver disengages the clutch, which causes the engine to speed up and the car to slow down, and on letting the clutch in again he gets either intermediate or low gear, depending on how much the engine speeded up and the car slowed down in the meantime.

This transmission, therefore, is selective; each gear can be used over the whole speed range of the engine and, if the engine is slowed down too much for smooth operation, a change to a lower gear is made automatically, thus preventing stalling.

An automatic shift for conventional three-speed-and-reverse transmissions has been invented by D. C. Prince of the General Electric Co.'s Philadelphia Works and has been applied by him to a Plymouth car provided with a vacuum-operated automatic clutch and a free-wheeling unit. Shifting of the gears is effected by means of a coiled spring which is first put under tension by a vacuum cylinder communicating with the inlet manifold of the engine through a valve. There is one floating lever for each of the slider bars of the transmission, the connection between the lever and the slider bar being of the same type as in a conventional transmission. Sliding fulcrums are used for this lever, and in one embodiment of the invention these fulcrums are controlled directly by a centrifugal device similar to a centrifugal governor. The sliding fulcrums are in the form of sliding fingers which can be either moved outward so as to bring them into the path of the lever, or withdrawn therefrom. There are two such sliding fulcrums for each of the slider bars, and the arrangement is as shown diagrammatically in Fig. 12. The lever is under the influence of the vacuum cylinder and the coiled spring, both acting on it at the point indicated by the arrow. The coiled spring pulls it in the direction indicated by the arrow, while the vacuum cylinder pulls in the opposite direction. The mechanism illustrated in Fig. 12 is that operating the high and second-speed slider bar. To engage high gear, the upper fulcrum finger is moved out, the lever finds a fulcrum on it, and the slider bar is pulled to the left by the coil spring, thereby engaging the direct drive. To engage the intermediate gear, the upper fulcrum finger is withdrawn and the lower fulcrum moved out, and the pull of the coil spring then engages second gear. The fulcrum fingers are moved automatically by the centrifugal device, which is under the influence of the car speed and of the accelerator pedal.

With the car at a standstill and the engine idling, the automatic friction clutch will be disengaged, owing to the high vacuum in the manifold. The various sliding fulcrums are "jammed" in their guides by the pressure on them of the springs which effect gear shifting. The driver now shifts the control lever to the "Forward" position. This closes an electrical contact and energizes a solenoid which in turn opens a valve in the line from the inlet manifold to the shifter cylinder. The piston of the shifter cylinder is drawn into it, thereby tensioning the shifter spring and releasing the fulcrum fingers, by drawing the shifter lever away from them. The fulcrum is now moved into a position (by a spring) where the shifter lever can engage it. At the same time the electrical contacts are opened so that the solenoid of the vacuum valve is de-energized, the piston in the shifter cylinder is released, and the spring pulls the low-speed gear into engagement. The pistons in the gearshift and the clutch vacuum cylinders are interconnected or interlocked so that the clutch can engage only very lightly before the gears mesh. This arrangement prevents clashing of the gears and also obviates failure to mesh when neither gear is rotating.

As the driver depresses the accelerator pedal, the clutch engages and the car starts in first gear. With further acceleration the centrifugal unit will move into a position where the slider bar for the intermediate and high speeds can be engaged. When this motion of the selector mechanism takes place the contacts of the valve solenoid are again closed, vacuum is admitted to the shifter cylinder, the movable fulcrums for the high-speed and intermediary slider bar are released, the fulcrum is pulled into its active position by a spring, and the shifter spring then shifts the second-speed gear into engagement.

Before the shift can occur, the main clutch must be disengaged and, as this disengagement is effected by vacuum in the inlet manifold, this vacuum must be increased by closing the throttle. For this reason the throttle disc is made free on its shaft so that the throttle can be closed without change in the position of the accelerator pedal. It is closed by a solenoid which is energized by the same electrical contacts as the solenoid of the valve in the vacuum line. When the contacts are opened again the throttle automatically opens to the position determined by the position of the accelerator pedal. The main clutch will reengage and the car will continue to accelerate, this time in second gear.

Other changes of gear are effected in the same manner, the change depending on the relative positions of a number of levers with notches in them. One of these levers is the hand control lever, which has three positions, Forward, Neutral, and Reverse. The other levers are controlled by the centrifugal device. It should be mentioned here that the speeds at which changes of gear occur are not fixed, but vary with the position of the accelerator pedal. The spring of the centrifugal governor is interconnected with the accelerator in such a

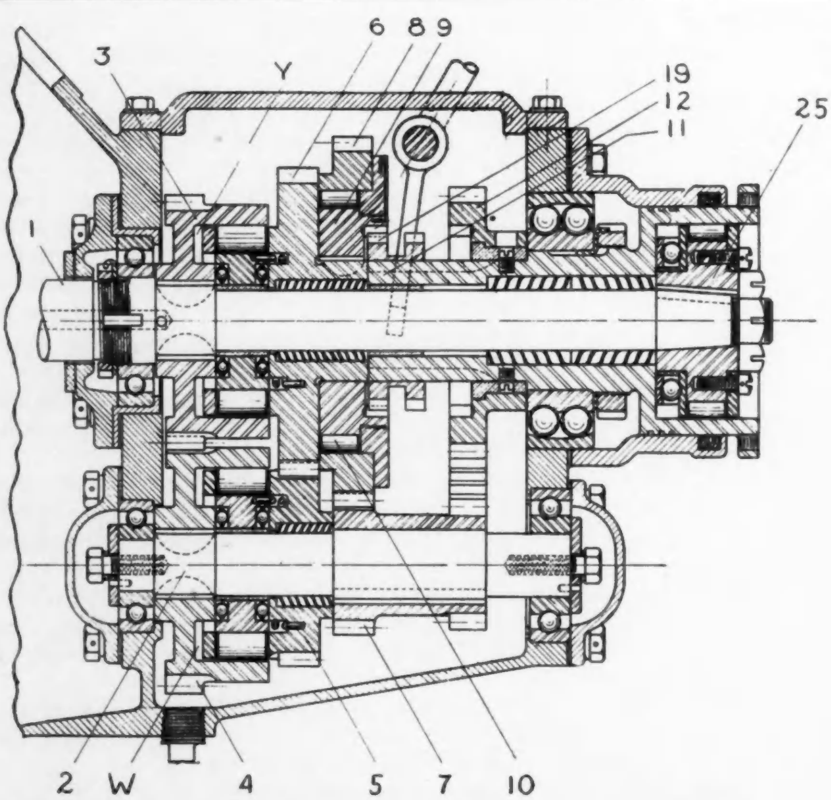


Fig. 11 – Longitudinal Section of Macallen Self-Shifting Transmission

way that, as the pedal is pressed down, the tension in the governor spring is increased, and any given upward shift is then effected at a higher car speed. That the driver presses the pedal farther down indicates that more driving torque is needed, and the engine can deliver this greater driving torque more effectively through the lower gears. For instance, the Plymouth to which this transmission was fitted developed maximum torque up to 20 m.p.h. in low gear and up to 45 m.p.h. in second gear, and these speeds were, therefore, selected for shifting at wide-open throttle whereas, for moderate throttle openings, it was found desirable to have the shift into high gear occur at 10-12 m.p.h. Conversely, when the car is slowing down on the level, it is not desirable to change to a lower gear until the speed has dropped to 8-10 m.p.h. whereas, on a hill the change from high to intermediate should occur at about 30 m.p.h. for best performance. These conditions are brought about automatically because, with the accelerator pedal depressed, maximum tension is exerted on the governor spring.

With this automatic shift, after the control lever has been set in the Forward position, speed control of the car is entirely by the accelerator pedal. There is no need for clutch operations under any conditions. In descending long hills, where it is desirable to use the engine as a brake, the gear can be locked. It is claimed that the mechanism is practically silent during changing operations and completely silent at all other times, and that it gives a selection of gears which closely approximates theoretical requirements.

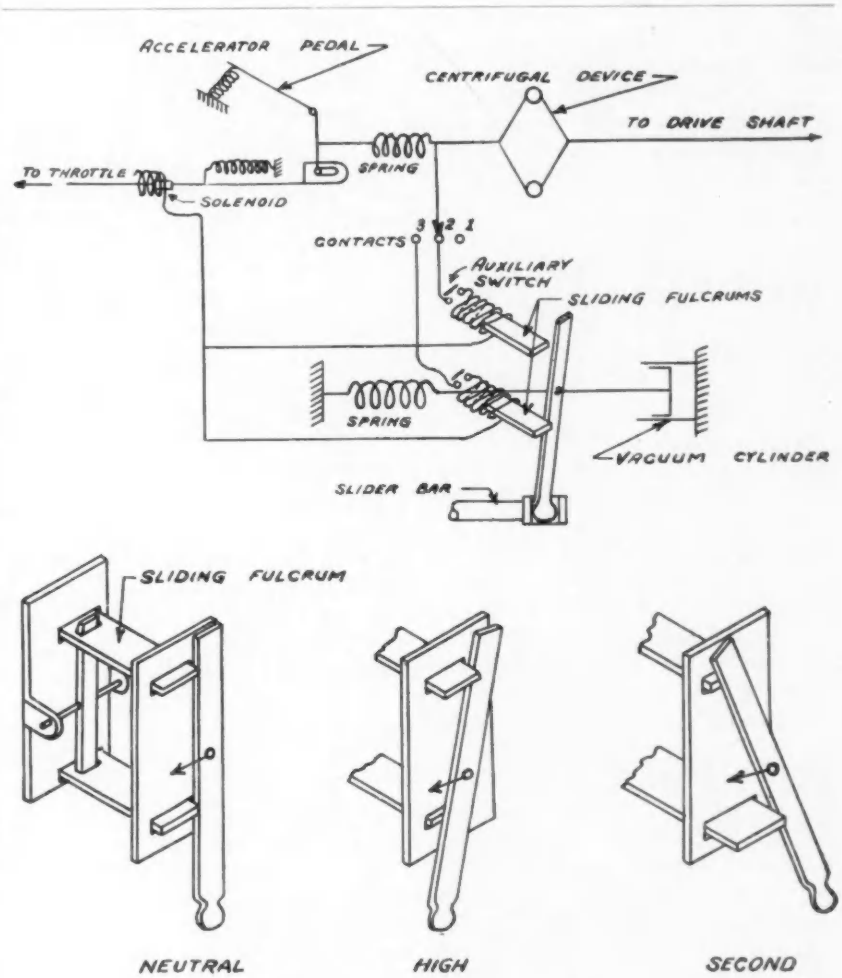


Fig. 12 – Diagrams Explaining Principles of Prince Self-Shifting Transmission

The Tyler Unimatic Motorcar Control, developed in Detroit, comprises a three-speed transmission of the constant-mesh type, the various forward speeds being engaged by means of individual multiple-disc clutches actuated by the inlet-manifold vacuum. In addition to the clutches for the individual gears there is a main clutch which transmits the power for all of the gears. This main clutch is engaged last (and disengaged first) so that the transmission clutches, which are smaller, are not subjected to the wear and tear of engagement under load. A cylinder-and-piston unit operates the main clutch, whereas the transmission clutches are operated by diaphragm chambers. Admission of vacuum from the inlet manifold to the cylinder and the diaphragm chambers is controlled by a sliding valve under the influence of a centrifugal governor driven from the transmission tail shaft at a speed proportional to car speed. There is a vacuum-control button adjacent to the accelerator pedal, by means of which the car can be free-wheeled when the accelerator is held in the idling position and, in addition, there is a master control button on the dash, which also serves to shut off the vacuum.

With the control button on the dash in the "automatic" position, as soon as the engine is started, the vacuum-control sliding valve shifts to the position corresponding to low-gear operation. However, the engine being throttled, the high vacuum in the inlet manifold does not allow the friction clutches to be engaged (by their springs) and the car therefore does not start yet. The driver now presses down on the accelerator, the engine speeds up, the inlet-manifold vacuum decreases and the clutches engage, so that the car starts. After a certain speed has been attained, the centrifugal governor shifts the sliding valve to the position for second speed. As the valve is held in its various positions by spring plungers, it snaps from one position to the next. However, to allow the shift to take place, the driver momentarily must release the accelerator, so that the driving members are relieved of torque. The high gear is engaged in the same way. The reverse gear is engaged manually, and all forward speeds also can be engaged manually, by means of the conventional shift lever, if the control button on the dash is set in the "non-automatic" position.

Besides the automatic friction clutch and the overrunning clutch, the differential gear has a place in automatic transmissions. A good many inventors have been intrigued by the possibilities of the differential gear for this purpose. Imagine an ordinary rear-axle assembly. If the crown gear is held stationary and one axle shaft is set in motion, the other axle shaft will turn at the same speed in the opposite direction. There is a tendency for the crown gear to turn in the same direction as the driving axle shaft and, if it is allowed to rotate slowly in that direction, then the driven axle shaft will rotate less rapidly in the reverse direction. If the crown gear is allowed to rotate at one-half the speed of the driving axle shaft, then the other axle shaft will be stationary and, if the crown gear is made to rotate still faster, then the second axle shaft will reverse its

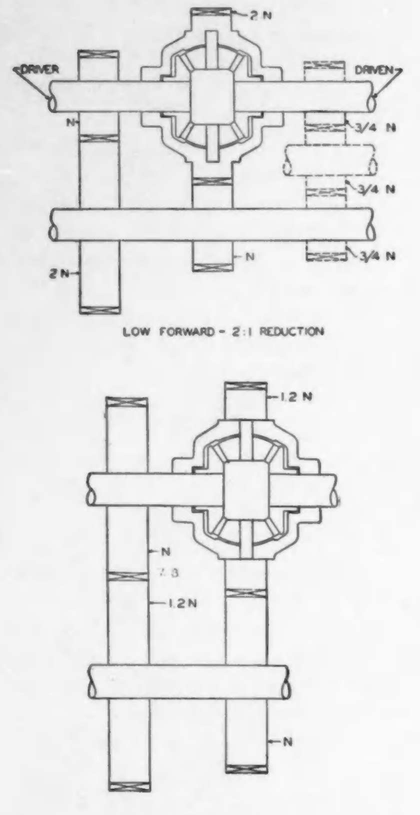


Fig. 13 – Diagrams Explaining Principle of Differential Transmission

direction of rotation. Thus it is possible, by controlling the speed of the crown gear, to cause the second axle shaft to pass through a wide speed range in both directions. In other words, any forward speed between the maximum and zero, as well as any reverse speed, can be obtained.

But how is the speed of the crown gear to be controlled? Many inventors have attempted to control it by friction, by applying a brake band to a drum connected to this gear. This method, however, is quite impractical, for it can be shown easily that, if the speed of the driven shaft is reduced by, say, one-half, then one-half of the power is wasted in overcoming friction. Besides, it would be quite impossible to assure accurate speed control with such a device for, if the frictional moment acting on the crown gear were less than the torque load on the driven axle shaft, the driven axle shaft would remain stationary and the crown gear would rotate at one-half the speed of the driving shaft. If the friction of the brake applied to the crown gear drum were now increased, as soon as the friction moment thus created exceeded that of the torque load, the crown gear would be stopped from rotation and the driven axle shaft would rotate at the same speed as the driving axle shaft (in the opposite direction).

In order to obtain a satisfactory degree of efficiency when transmitting power with a device of the type in question, it is necessary that the power absorbed by the crown gear when turning under the torque impressed on it, be returned to the transmission line. This return can be accomplished by either mechanical, hydraulic, electric, or pneumatic means, and the power may be returned from the crown gear to the driving shaft or it may be transmitted from the crown gear directly to the driven shaft. Return of power from the crown gear to the transmission line by positive mechanical means is not practicable because of the impossibility of an infinite variation of the gear ratio, but this method lends itself well to an illustration of the properties of the differential gear as a torque converter.

In Fig. 13 the diagram at the top shows a gear combination which gives a speed reduction of 2:1. The output shaft turns in the opposite direction to the input shaft, but this may be considered a forward drive. In this case the input shaft is connected to the crown gear by a train of gears which gives a reduction of 4:1, so that the crown gear turns in the same direction as the input shaft, but at one-fourth its speed. Power absorbed by the crown gear is returned to the input shaft. Instead of connecting the crown gear to the input shaft by gearing, we could connect it to the output shaft, as indicated in dotted lines, using an intermediate pinion, because the output shaft turns in the reverse direction to the crown gear. Also, as the output shaft turns at only one-half the speed of the input shaft, the gear reduction from the output shaft to the crown gear must be 2:1 instead of 4:1.

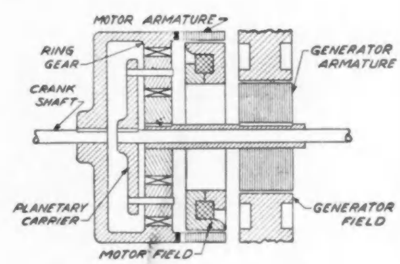
It is obvious that if we used either of these two arrangements and made the output shaft the input shaft, we should have an overdrive with a ratio of 1:2.

For a reverse drive it is necessary to make the crown gear turn faster than one-half the speed of the input shaft and in the same direction, which can be done by providing a gear train of the proportions shown on the lower diagram in Fig. 13.

The differential gear actually used in such transmissions is not the bevel-gear type commonly used in rear-axle drives, but a planetary combination consisting of a sun gear or pinion, a number of planetary pinions supported on studs extending from a planetary carrier, and a ring gear with internal teeth. This type of gear is free from end thrust, which would be rather bothersome if the ordinary bevel-type differential were used.

A differential transmission of the electric type is illustrated diagrammatically in Fig. 14. Here the ring gear is secured rigidly to the engine crankshaft and meshes with a number of planetary pinions supported on a planetary carrier mounted on the driven shaft. The latter is surrounded by a hollow shaft which carries the armature of an electric generator and, between the planetary assembly and the generator, there is an electric motor whose armature is integral with the ring gear.

Fig. 14 – Diagram Explaining Principle of Electric Differential Transmission



The driven shaft is in driving connection with the car and, therefore, under load. When the engine is started up, the ring gear turns with it, the planetary carrier is held from rotation by its load, and the sun pinion, therefore, turns in the direction opposite to that of the engine. With it the generator armature revolves and, as a result, a current is generated in it. This current is sent through the electric motor and, as the motor armature is carried by the ring gear, the motor torque is added to the engine torque. The combined torque will start the car, thus setting the planetary carrier in rotation, and the speed of the generator relative to that of the engine will then be reduced. At steady car speeds (when there is no acceleration) on the level the engine torque is sufficient to propel the car through the rear-axle gearing, and the planetary carrier will then rotate almost as fast as the crankshaft, so that the sun pinion and the generator armature rotate at only very low speeds. The electromotive force then generated is very low, and little of the power

of the engine is converted into electric power, the greater part of it being transmitted directly. With a drive of this kind the efficiency, therefore, is higher than in a straight electric drive where all of the power is converted into electric power in the generator and then converted back to mechanical power in the motor. With such a drive, speed control is by means of the engine throttle.

The electric generator in Fig. 14 could be replaced by some kind of pump and the electric motor by a hydraulic motor, and the principle would be essentially the same. Such differential hydraulic transmissions of both the hydrostatic and hydrodynamic type have been worked out. A hydrostatic differential transmission, based on the Waterbury hydraulic drive, is shown in longitudinal section in Fig. 15. It consists of the two Waterbury units, the pump and the motor, with a valve plate and a differential between them. The shafts of the two units, which in the conventional design of the Waterbury hydraulic transmission (Fig. 4) are independent of each other but have bearings in the central block, in this design are secured to the side gears of a bevel-type differential.

There is also another kind of transmission making use of the "feed-back" principle, but without the differential gear. In this case the differential action is produced by a magnetic or hydraulic coupling. The best example of this type is the Entz transmission, which was used for a number of years on the Owen Magnetic car. A sectional view of this transmission is shown in Fig. 16. It comprises two electrical machines arranged on the same axis, the one nearest the engine being referred to as a generator and the other as a motor. The field frame of the machine next to the engine is bolted to the flywheel flange of the engine and serves as its flywheel. The armatures of both machines are secured to a large tubular shaft mounted in ball bearings at opposite ends. This is really the driven shaft of the transmission, but a sliding type of reversing gear is incorporated between it and the propeller shaft.

When the engine is idling, the generator field, of course, revolves with it, but as the generator circuit is open there is no drag on the armature and there is therefore no torque on the driven shaft. The controller is then moved to the starting position, which connects the electric motor to the generator. Owing to the relative motion between the generator field and armature, an electromotive force is induced in the generator armature and, as a result, a current flows through the motor. This current causes a driving torque to be produced by both the generator and the motor. The magnetic drag between the generator field and armature is exactly equal to the engine torque, and this torque, therefore, is impressed on the hollow propeller shaft. The electric motor produces a torque of its own, and this also is impressed on the hollow driven shaft. As long as the car speed is still low there will be a large slip between the generator field and armature, with the result that a large electromotive force is generated, the motor receives a large current and produces a heavy torque, which is good for acceleration. As the car gains speed, the slip of the generator armature decreases and with it the electromotive force generated and the torque produced by the motor.

There are a number of different controller positions and, in the final or high-speed position, the electrical connection between the generator and motor is broken and the generator is short-circuited upon itself. It then acts as a magnetic coupling, transmitting the torque of the engine directly to the driven shaft, with a slippage of only a few per cent, depending on the load.

An electro-mechanical differential transmission has been developed in this country and is to be marketed under the name Electrogear. With this transmission, on starting from a standstill with the propeller shaft stationary, when the throttle is suddenly opened wide for quick acceleration, nearly all of the power generated is converted into electrical power, whereas under normal driving conditions the proportion so converted (and reconverted into mechanical power), varies between 5 and 30 per cent.

The Electrogear (Fig. 17) is built in the form of a unit that is flanged to the rear of the engine block and has the propeller shaft connected to its output shaft through a universal joint. It comprises a booster dynamo driven directly by the engine, a differential gear of the planetary type, a reducer dynamo whose armature is connected to the engine crankshaft and the output shaft through the differential gear, an electrical circuit including the two dynamos and a control switch, and a regulator dynamo whose current is used to vary the field excitation of the booster dynamo. A sectional assembly view is shown in Fig. 18.

It can be seen that the armature of the booster dynamo is built on a cylindrical shell or drum, within which is located the differential gear. The latter is of the planetary type, con-

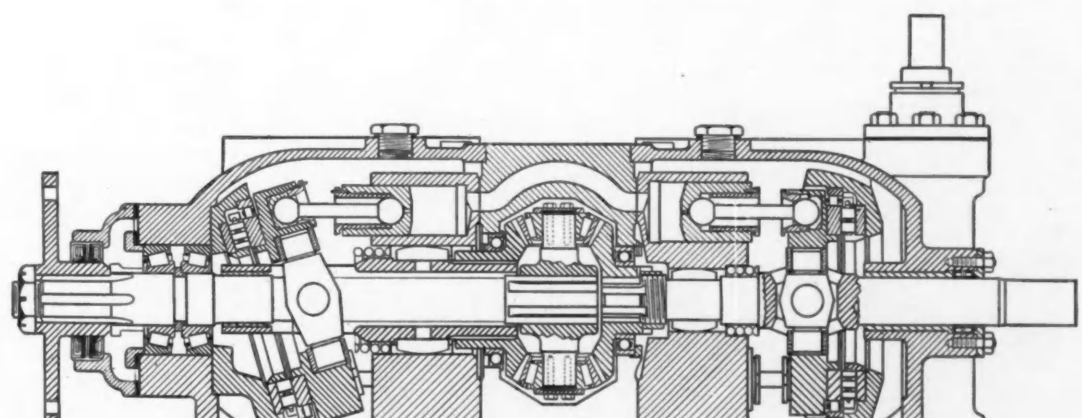


Fig. 15 – Hydrostatic Differential Transmission

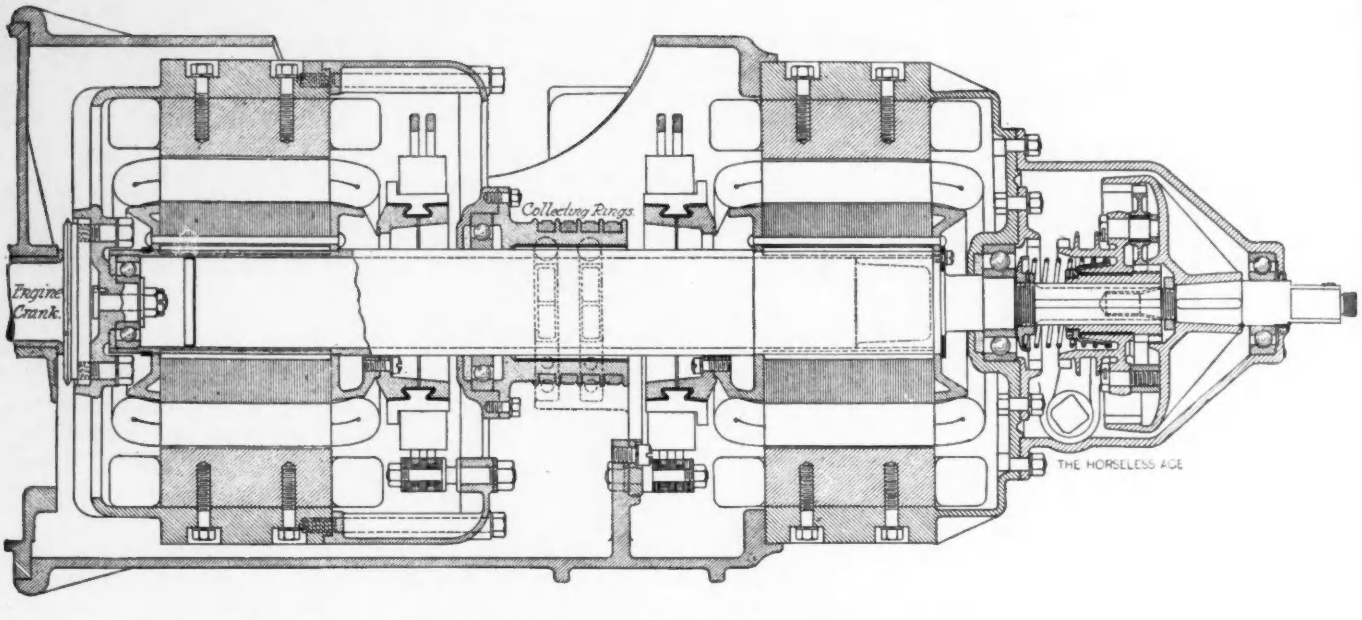


Fig. 16 – Entz Electric Transmission of Owen Magnetic Car

sisting of a sun pinion, three planetary pinions, and an internally-toothed ring gear, all of them cut with helical teeth. Both the ring gear and the armature of the booster dynamo are fastened to the crankshaft and rotate with it. These parts, moreover, serve as flywheel for the engine. At its rear end the booster armature is supported by a ball bearing mounted in a bulkhead of the transmission housing. The planetary carrier is splined to the forward end of the output shaft of the transmission, whereas the sun pinion has a hub or lateral extension that is splined to a tubular shaft carrying the armature of the reducer dynamo.

A car equipped with the Electrogear is controlled by means of a switch on the dashboard and the accelerator pedal. The switch, which needs to be manipulated only with the car at rest, has four positions, namely, "Cranking," "Forward," "Neutral," and "Reverse." When the switch is turned to the "Cranking" position, the booster dynamo is connected to the car battery and acts as a starter motor, cranking the engine over. To warm the engine up, the switch is turned to "Neutral." While this operation is being done, no driving torque is exerted on the car wheels because the dynamo circuit is still open. Next the driver turns the switch to "Forward" and then depresses the accelerator pedal, the first motion of which closes the circuit. Further depression opens the throttle and speeds up the engine.

The planetary carrier being in driving connection with the driving wheels of the car, there is considerable resistance to its rotation and, when the engine is first speeded up, rotation of the ring gear (with the crankshaft) results in rotation in the opposite direction of the sun pinion and the reducer armature connected to it. Current generated by the reducer dynamo passes through the booster dynamo, which now acts as an electric motor, generating driving torque which is added to that of the engine. The current generated by the reducer dynamo and the torque produced by the booster dynamo increase and decrease with the speed of the reducer armature, as does also the reactive force generated by the reducer dynamo and impressed on the sun pinion of the differential. As a result of this reactive force, torque is transmitted from the ring gear to the planetary carrier, and the car is accelerated. A multiplication of the torque is effected by the gearing.

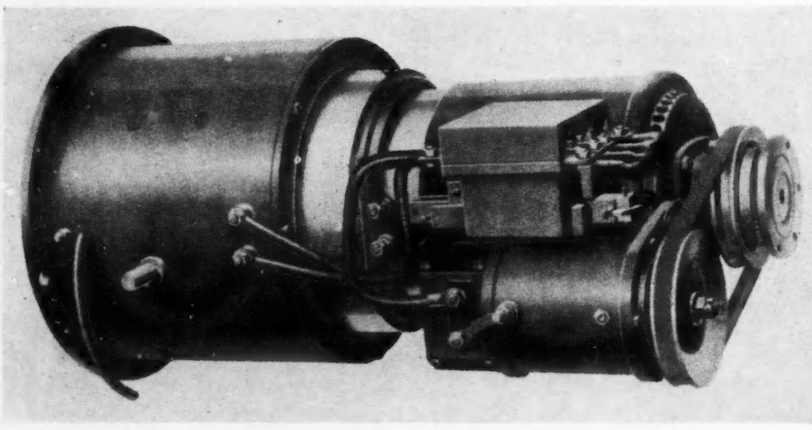


Fig. 17 – The Electrogear Unit

The object of the regulator dynamo is to modify the magnetic field of the booster dynamo. The electromotive force generated by the regulator dynamo is impressed on the field winding of the booster dynamo and acts in opposition to the electromotive force normally impressed on this field, which originates in the reducer dynamo. As the regulator dynamo is driven at a speed proportional to the car speed, its electromotive force increases with car speed. Therefore, the effect of the electromotive force generated by the regulator dynamo is to first weaken the booster dynamo field and later, at higher car speeds, to reverse that field and convert the booster from a motor to a generator. Thus, with increasing car speed, more and more of the driving torque is supplied directly

by the engine, so that the latter is prevented from attaining excessive speeds. This trend continues until the booster no longer supplies torque to the differential and the reducer armature has almost come to a stop from its rotation in the reverse direction. Under these conditions less than 5 per cent of the engine power is converted into electric power. The torque on the output shaft is now 1.4 times that on the crankshaft (for the proportions shown in the drawing).

At low speeds, owing to the combination of booster torque with crankshaft torque, the torque-multiplication ratio is considerably greater, of course.

As the car speed increases further, the regulator dynamo reverses the field of the booster dynamo and makes a generator out of it. Current from the booster dynamo then passes through the reducer dynamo which, acting as an electric motor, starts to turn in the opposite direction. With both the ring gear and sun pinion rotating in the same direction, the ratio of crankshaft speed to output-shaft speed (the reduction ratio) decreases further and, when the reducer armature has attained a speed equal to that of the crankshaft, the differential rotates as a solid unit and the drive is direct. Under this condition the booster dynamo, acting as a generator, imposes a considerable load on the engine and, therefore, pulls it down in speed. However, the booster dynamo continues to build up as a generator, thus further speeding up the reducer armature and the output shaft through the differential gear, until the propeller shaft turns at 50 per cent or more above crankshaft speed, giving an overdrive. One feature of this transmission is that the electrical units can be proportioned readily so that at maximum car speed the engine is operating at the peak of its power curve whereas, in a conventional car with the stepped transmission, it usually operates considerably beyond this point.

In normal operation all speed control is exclusively by the accelerator pedal. As the driver releases the pedal, the torque on the output shaft and the current passing between the two dynamos are reduced. This action immediately weakens the reducer field but, owing to the reduction in the speed and voltage of the regulator dynamo, the booster field is strengthened and the additional engine load due to the booster acting as generator holds the engine down to a moderate, uniform speed.

To reverse the direction of the drive it is necessary that the reducer armature turn in the opposite direction to the crankshaft and that the pitchline velocity of the sun pinion be greater than that of the ring gear, which means that the reducer armature must turn several times as fast as the crankshaft. In reverse, the booster dynamo acts as a generator and sends a current through the reducer dynamo in such a direction that the latter turns in the opposite direction to the booster armature.

With this transmission there is a continuous variation in the torque ratio, which takes place automatically in accordance with the driving torque required and the accelerator position. There is no interruption in the driving torque when the speed ratio is changing.

Under light load an overdrive is obtained, the driven shaft of the transmission then turning faster than the crankshaft. Maximum overdrive ranges from about 30 per cent at 30 m.p.h. to 50 per cent at 50 m.p.h. This overdrive feature may result in a gain in economy as compared with a car with the conventional transmission without overdrive.

As designed for use on private cars, the Electrogear weighs about 5 lb. per hp., 90 per cent of this weight being accounted for by the electrical and 10 per cent by the mechanical units. This is about twice as much as the weight of the parts eliminated.

A hydrodynamic type of differential transmission has been produced by Bendix Products Corp., and is known as the Bendix Turbo Flywheel Gear. As illustrated by the sectional view in Fig. 19, it comprises a hydraulic torque converter whose impeller or driving member forms a housing that also serves as the flywheel of the engine; a differential gear of the plane-

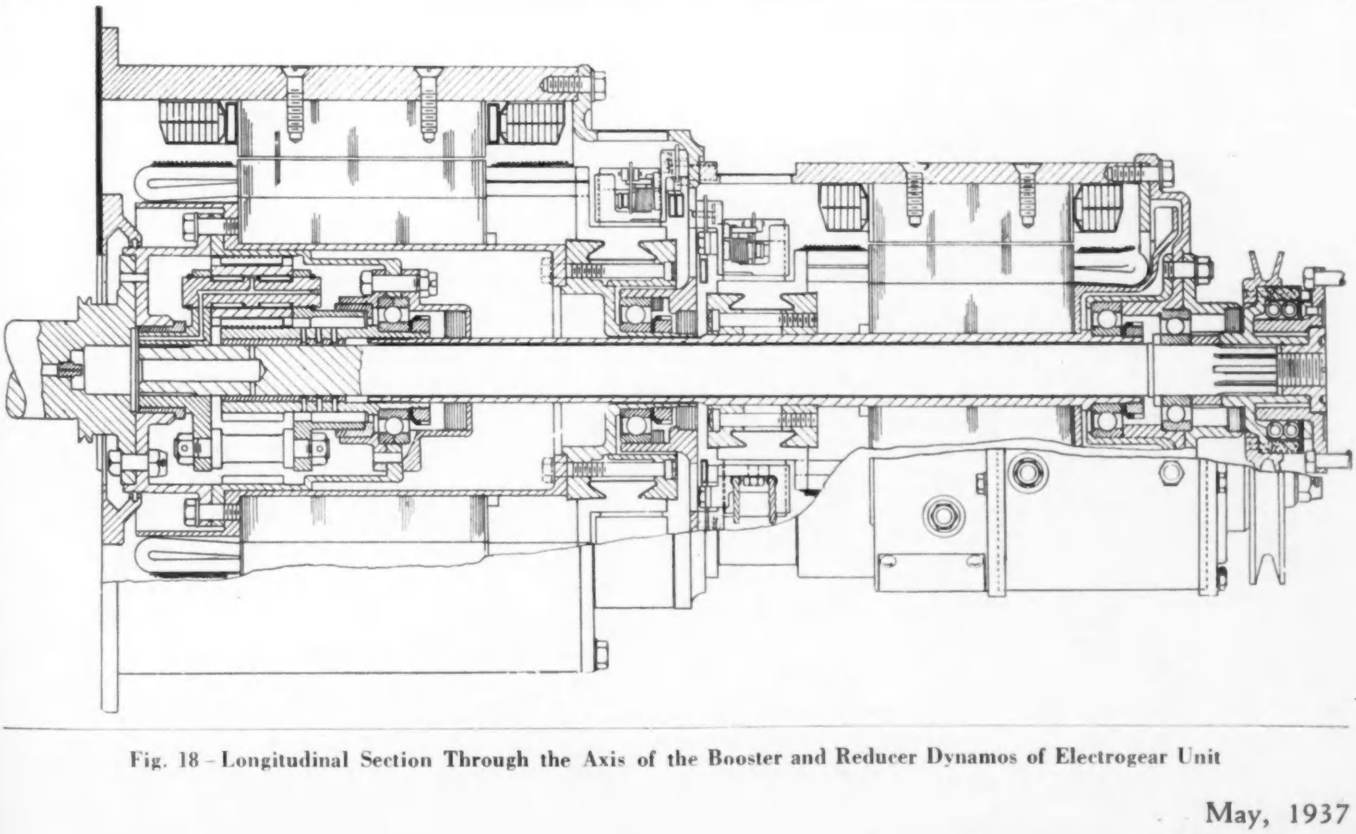


Fig. 18 – Longitudinal Section Through the Axis of the Booster and Reducer Dynamos of Electrogear Unit

tary type; and shifting mechanism by means of which the connections between the elements of the differential gear and the input and output shafts can be changed.

Control of this transmission is by means of a shift lever which has four positions, namely, "Reverse," "Neutral," "Forward," and "Forward Emergency Low." Shifting is done only with the car at rest. The "Emergency Low" is practically never required for driving under power, and is provided mainly as a safety feature for use on treacherous slippery down grades.

Referring to the sectional drawing of the assembly (Fig. 19) in the hydraulic unit shown at the left, *I* is the impeller, R_1, the first rotor, *S*, the reaction member, and R_2, the second rotor. The impeller *I* is part of the flywheel; both rotors are secured to a tubular shaft or sleeve, while the reaction member *S* is mounted on a stationary hub extending inward from the rear portion of the housing in such a manner that it is free to turn around its housing in one direction, but not in the other.

A central driveshaft extends from one end of the transmission to the other, and the whole mechanism is virtually carried on this shaft. In the regular forward drive this central shaft *A* is coupled to the flywheel (or crankshaft) by means of clutch teeth upon it and in the bore of the flywheel center, and then turns at engine speed. Ring gear *B* is connected to the hub of rotor R_1R_2 by a toothed clutch between the hydraulic unit and the planetary assembly, and planetary carrier *C* is connected to output shaft *E* by a toothed clutch within the drum-shaped inner end of the shaft. Both the sun pinion and the ring gear turn in the same direction, and they carry the planetaries and the carrier *C* along with them. When the drive is direct, the sun pinion and ring gear turn at substantially the same speed, and they then contribute to the power transmitted in proportion to their pitchline velocities, which are proportional to their pitch diameters (1:3 in the drawing). At the start, of course, there is considerable slippage in the hydraulic device; the ring gear then turns at lower speed than the sun pinion, and the proportion of the power transmitted by the hydraulic unit is then relatively less.

In "Emergency Low," ring gear *B* is locked to the housing by the toothed clutch between the hydraulic unit and the planetary assembly, and it therefore remains stationary. Planetary carrier *C* remains connected to output shaft *E* and sun pinion *A* is connected to the hydraulic rotor by the clutch teeth on shaft *A*. Thus the power is transmitted first through the hydraulic unit and then through the planetary assembly, which latter, with the sun pinion as the driving, and the planetary carrier as the driven member, gives a reduction ratio of 4:1.

In "Reverse" the sun pinion remains connected to the hydraulic rotor, the planetary carrier is now locked to the housing by the toothed clutch between the hydraulic unit and the planetary assembly, and the ring gear is clutched to the output shaft. In this case, with the planetary carrier stationary, the ring gear turns in opposition to the sun pinion and the output shaft in opposition to the crankshaft. With the proportions shown, the reduction ratio for reverse drive is 3:1.

With the transmission as illustrated herewith there is always at least a slight slip between the impeller and rotor of the hydraulic unit, but a positive direct drive, which eliminates this slip, can be incorporated in it. An efficiency curve for this transmission shows that for constant car speed (no acceleration) the efficiency is more than 90 per cent at 20 m.p.h. and reaches approximately 95 per cent at 80 m.p.h. This high efficiency is explained by the makers on the grounds that in contrast to conditions in ordinary hydraulic turbines, any energy left in the fluid after it has passed the rotor blades is not lost with water discharged, but is returned to the impeller blades; secondly, there is no loss due to friction of the fluid passing over stationary housing walls and guide vanes, as the casing turns at impeller speed and the guide vanes are fixed only a small part of the time.

Although this transmission has no overdrive and the output shaft never rotates faster than the input shaft, it possesses the same economy feature as an overdrive, for on account of the wide range in the torque ratio obtainable, a smaller rear-axle reduction ratio can be used. In addition to increased fuel economy, an increase in the acceleration is claimed, and the hydraulic unit eliminates shock on the transmission members.

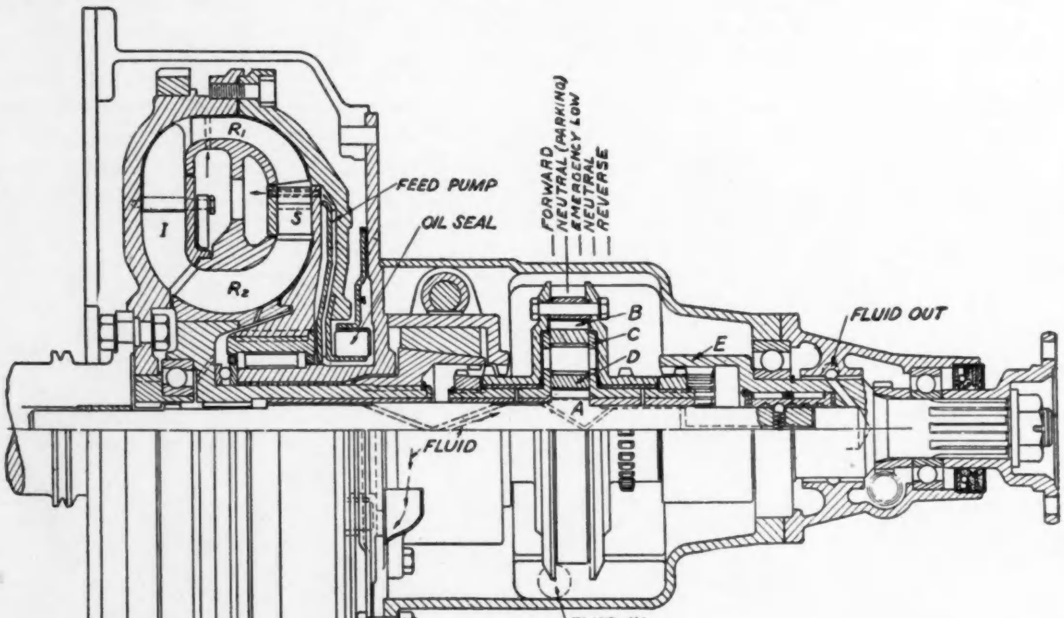


Fig. 19 – Longitudinal Section of Bendix Turbo Flywheel Gear